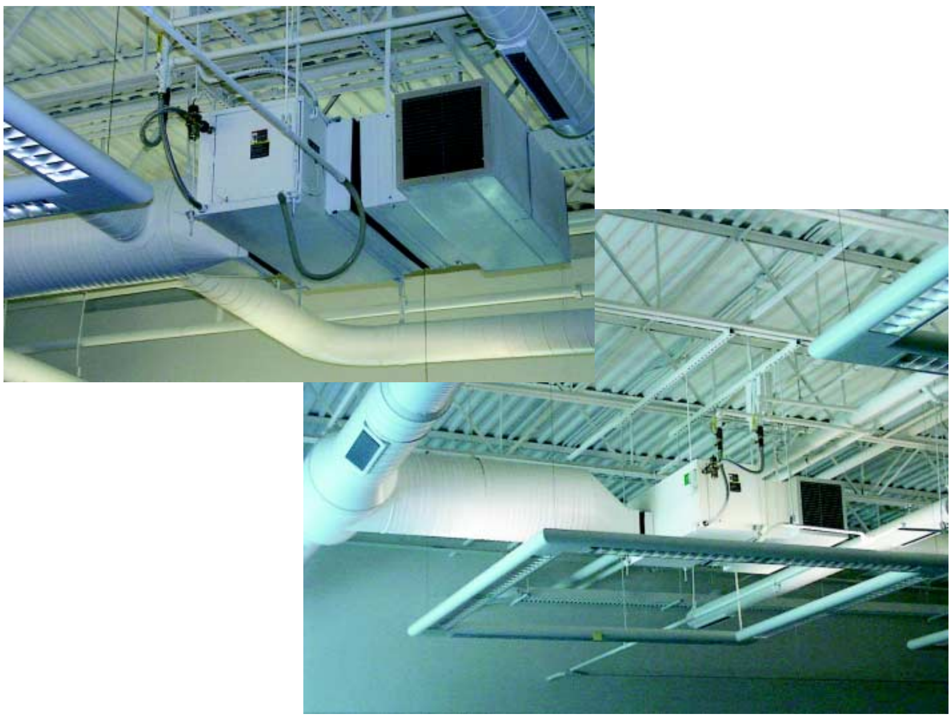
Catalog C: 330-1

Water Source Heat Pump Design Manual

A Design Manual for the Professional Engineer



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(5/99)

The decentralized approach

Multi-zone or multi-room buildings have two characteristics whose importance HVAC system engineers too frequently underestimate: diversity and seasonal loads (see page 4).

Diversity can be defined as the non-occurrence of part of the cooling load. The probability of having all the people present in the building, all lights operating (or operable) and all heat producing equipment operating at the time of peak design load is slight with lower probability on larger buildings. Most engineers allow for diversity of cooling load by selecting equipment with a cooling capacity smaller than the maximum potential cooling load. The degree of difference is strictly a judgment factor. If the engineer guesses wrong, or the usage pattern of the building changes, the cooling system may wind up oversized or inadequate.

Nearly all central HVAC systems have poor part-load efficiencies. At design load conditions, the best central systems operate magnificently, but during most of the annual operating hours, they consume a disproportionate amount of energy, maintaining a holding pattern, contributing very little energy to actual building heating or cooling.

The desirability of having heating or cooling available in any room, at any time, is obvious, but most central systems fill this need with “energy bucking” approaches, which divide the air conditioning medium (air or water) in two; part is overheated and part is overcooled. The medium is delivered to the space, mixing the hot and cold quantities as required to maintain the desired space temperatures.

Other systems are energy neutral, and newer versions have been falsely touted as energy conservation systems. Compared with their energy wasting predecessors, they represent a considerable advancement in the state of the art, but they do not actually store surplus energy for later use.

The first major step in reducing the annual power consumption for a multi-zone or multi-room building is to abandon the central system approach in functional areas where it has been demonstrated unfit — the heating and cooling of the various rooms. Present technology still mandates central sprinkler systems, and probably fresh air ventilation systems. For the heating and cooling functions, however, **a terminal unit in each zone or room provides inherent energy conservation**. Each unit heats or cools as required, whenever desired, only to the extent necessary, thus allowing the realization of diversity in heating, cooling and electrical use.

The second major step is to make the terminal units water source heat pumps, and interconnect them with a closed water loop. This allows transfer of energy from satisfied spaces in the building to areas lacking sufficient energy. The closed water loop permits efficient energy transfer (there is probably no *less* efficient method of transferring energy over long distances than using air as a heat transfer medium).

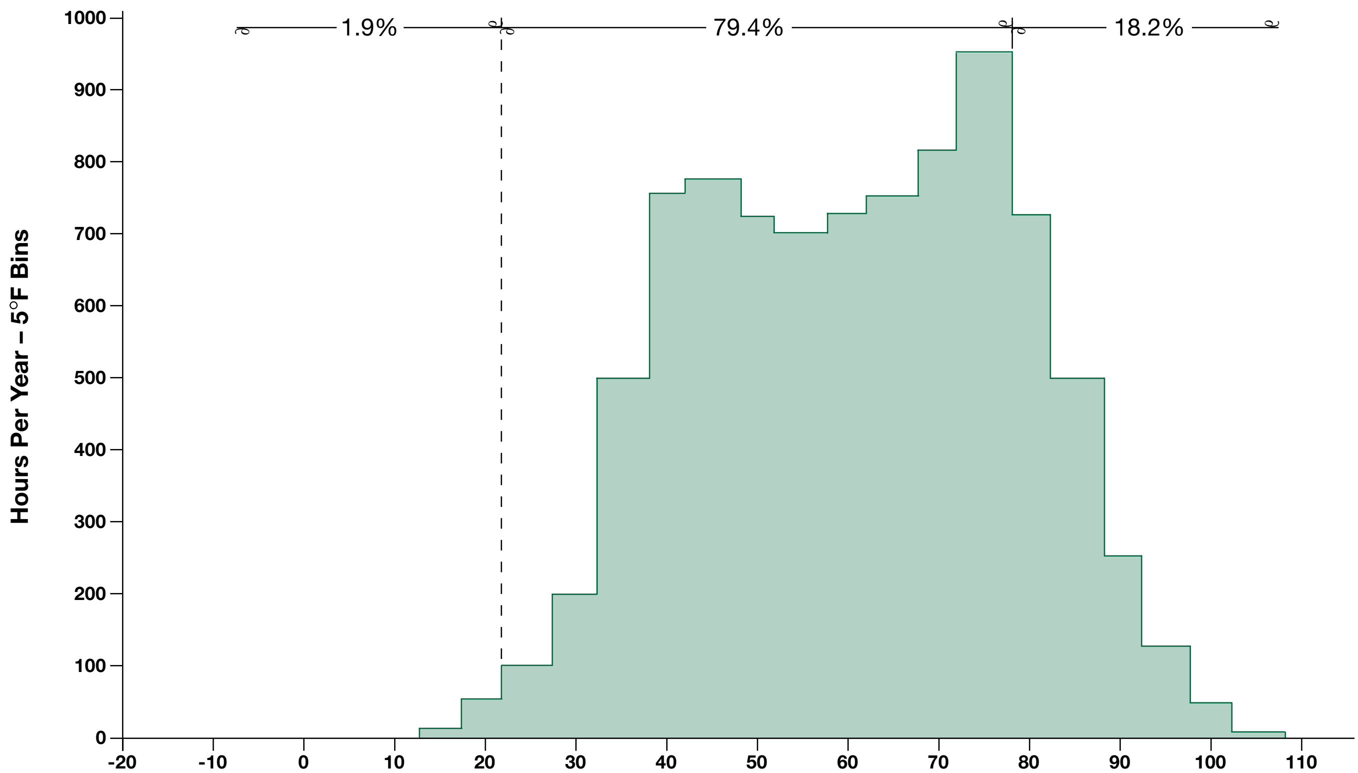
The Closed Water Loop Heat Pump System has gained wide acceptance among owners and designers to the point where it is the preferred system for multi-room or multi-zone buildings. For example, a U.S. Department of Defense directive states, “The most efficient method of using electric power for heating is the water source heat pump… Accordingly, when consideration is being given to the use of heat pumps, the water source should be evaluated first… the air source heat pump is second choice”.[1] Unlike most unitary heat pump systems, the closed loop system is employed to the greatest advantage in cold weather climates typified by Toronto, Minneapolis, Syracuse, or Milwaukee.

Among the many benefits realized with a water source heat pump system are:

- Ultimate flexibility of zoning.
- Maximum diversity at all times; the units only operate as required by their individual space controls.
- Ability to heat one zone and cool an adjacent zone.
- Smaller mechanical rooms because no large central refrigeration equipment is required.
- Building volume is decreased, or usable space is increased, because ductwork is minimal and basic energy transmission occurs through electrical wiring and uninsulated pipes.
- Less field labor required to install than with built-up systems.
- Simplicity of design. No complicated control valves or extensive, field erected automatic temperature control system.
- Maximum system reliability. A failure in one unit does not affect the others.
- No necessity to employ a licensed equipment operator, and less expensive maintenance contracts, as any unit may be removed, replaced by a spare, and the unit returned to a local repair depot for repair, and later returned to the building for use as a spare.
- Maximum architectural design flexibility, both in basic building configuration and interior layout. Terminal units are available in under-window console models, ceiling concealed or vertical closet types, large package units in capacities to 30 tons (105 kW), and rooftop units where there is no interior place to install the equipment.
- Minimum initial investment for speculative type office or apartment buildings, as the water loop can be designed and installed without prior knowledge of the ultimate floor arrangement within the leased areas, and terminal equipment can be later purchased and installed as required.
- Basically a constant year-round electrical demand, with any supplementary heating requirement obtainable on an off peak basis by limiting the demand, or through the use of a load control which enables the water temperature to be increased during periods of off peak demand.

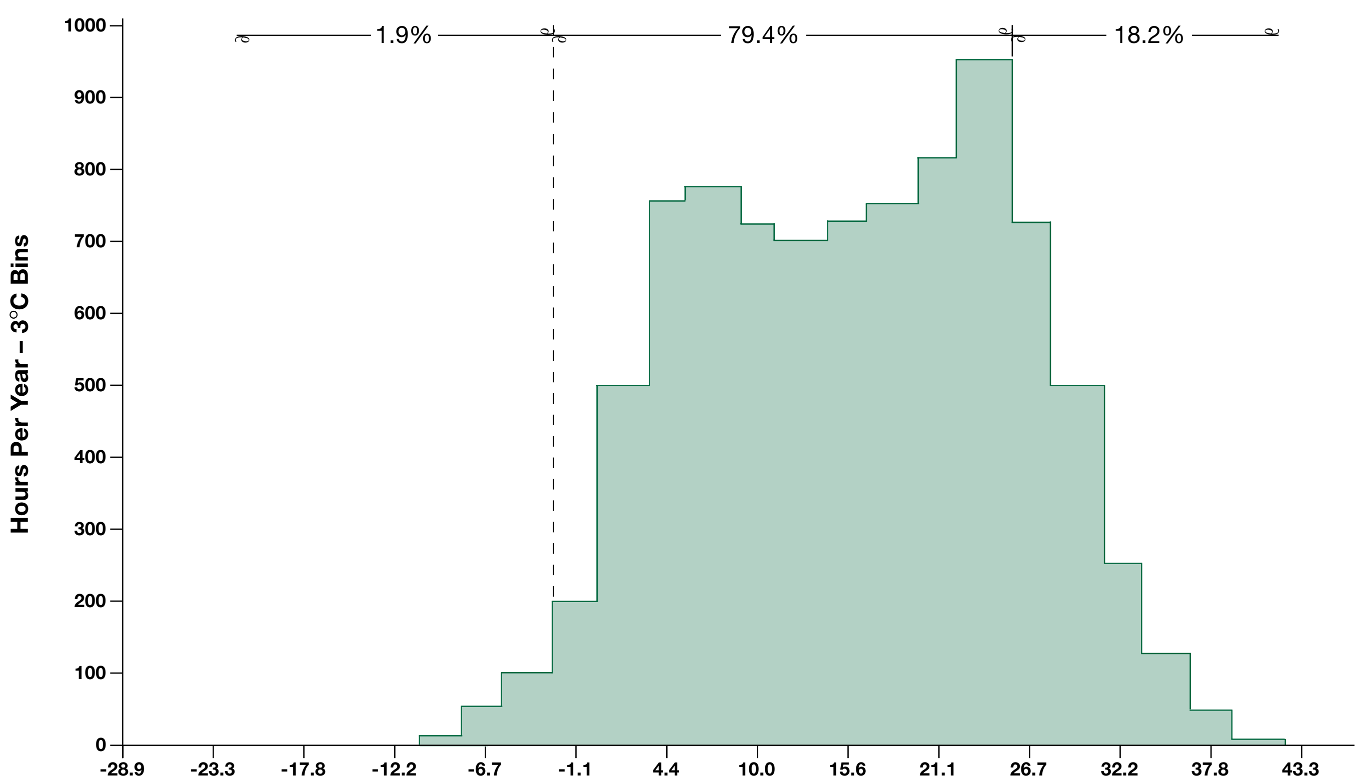
[1] DOD memorandum, May 12, 1975, Perry J. Fliakas, Deputy Secretary for Installations and Housing

Temperature Occurrence Profile – Washington, D.C.



Outside Temperature (°F)
4224 Degree Days

Temperature Occurrence Profile – Washington, D.C.



Outside Temperature (°C)
4224 Degree Days

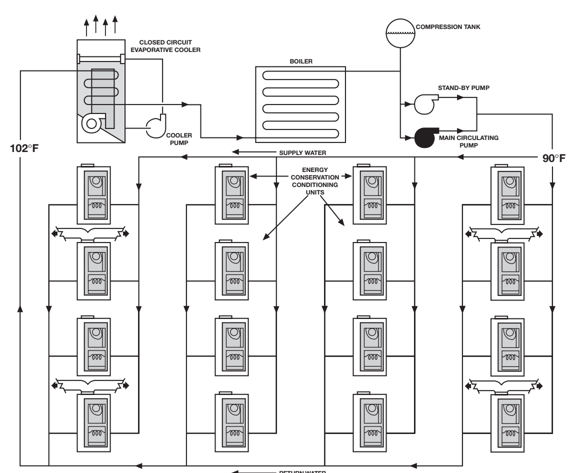
Chapter 1. Design steps

A. System description

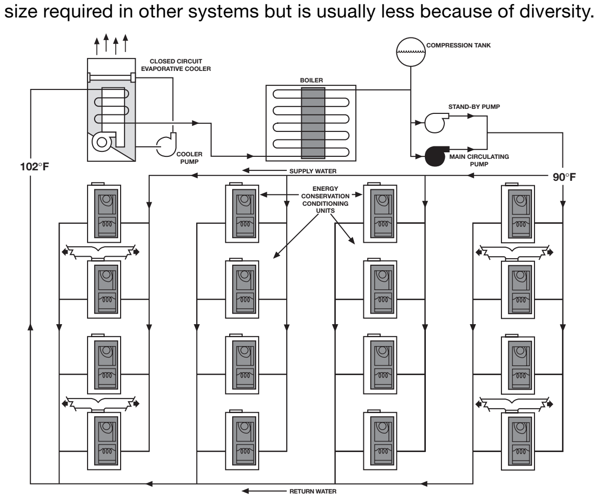
This decentralized, year-round heating and cooling system consists of a two-pipe closed loop water circuit, through which non-refrigerated water circulates continuously throughout the building. Locating the piping within the building negates the need for piping insulation. A supplemental central heat source adding heat to the loop at the lower end of the range and heat rejecter equipment capable of removing heat at the high end of the range maintains the loop water temperature throughout the year in an approximate range of 65° to 95°F (18.3° to 35°C). Filled with water, this circuit provides both a "sink" and "source" of energy. These systems achieve energy conservation by pumping heat from warm to cold spaces whenever they coexist anywhere within the building.

On demand for heating a space, the conditioner will absorb heat *from* the loop circuit, whereas on demand for cooling a space, the conditioner will reject heat *to* the loop circuit. The system provides the essential benefit of decentralized and individual choice of heating or cooling… the occupant may select heating or cooling or may shut off the conditioner serving an individual space without affecting conditions maintained in other spaces. The occupant may realize this freedom at any time of the day or year.

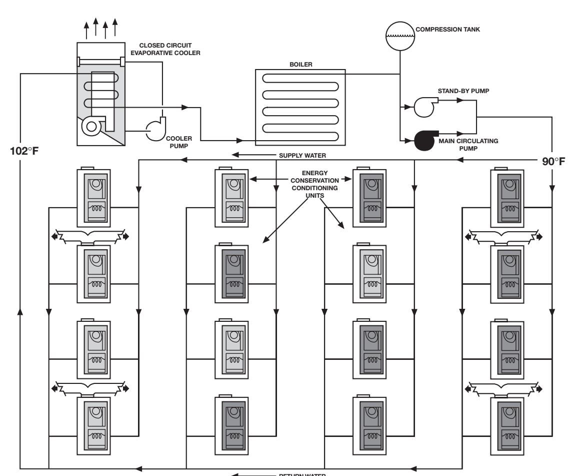
1 During hot weather with most or all units cooling, heat removed from the air is transferred to the water loop. An evaporative water cooler rejects the excess heat outdoors to maintain a maximum water temperature of approximately 90°F (32°C).



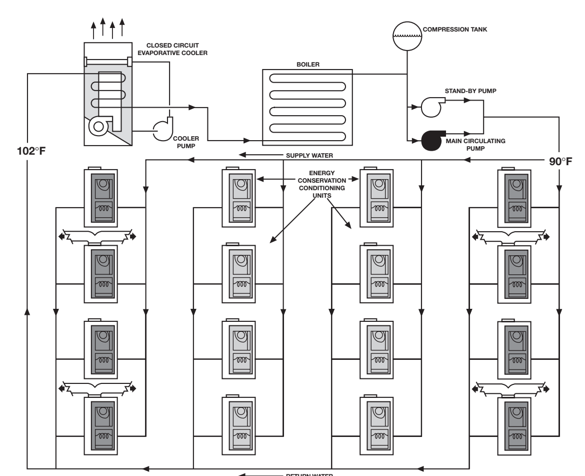
2 Only in very cold weather with most or all units heating is it necessary to add heat to the water with a water heater. This is done when the temperature of the water loop falls to 64°F (18°C). The amount of this heat is reduced any time one or more units are operating on cooling. The central water heater is never larger than two-thirds the size required in other systems but is usually less because of diversity.



3 In moderate weather, units serving the shady side of a building are often heating while those serving the sunny side require cooling. When approximately one-third of the units in operation are cooling, they add sufficient heat to the water loop so that neither addition to nor rejection of heat from the water is required.



4 Applications such as office buildings with high heat gain from lights, people or equipment in interior areas may require cooling of the space year-round. Heat taken from those areas is rejected to the water loop providing enough heat for the building perimeter any time at least one-third of the air conditioners' capacity is operating on cooling.



Units on cooling   Units on heating

B. Establish block cooling load of building

This should be calculated by the methods shown in the ASHRAE "Handbook of Fundamentals." Enter block cooling load on design worksheet. A sample worksheet appears in Chapter 8.

C. Establish block heating load of building

This should be calculated by the methods shown in the ASHRAE "Handbook of Fundamentals." Enter block heating load on design worksheet.

D. Select all units for building

After computing all heat losses and gains, select terminal heating and cooling units for each room or zone in the building.

1. **Conventional system** — Select for the greater of the heating or cooling load. Base selection on unit cooling capacities with 100°F (37.8°C) leaving water temperature, or on unit heating capacities with 65°F (18.3°C) leaving water temperature.
2. **Boilerless system** — Select unit for cooling load only, and specify electric resistance heater of sufficient capacity to offset room heat loss.
3. **Either system** — Take advantage of any opportunities to use subzone coil condenser water reheat (cooling) or recool (heating) providing that the subzone has a lesser heating and cooling requirement than the primary zone.

E. Select the evaporative water cooler

1. Summarize the total rated cooling capacity of all terminal units in Btuh or kW. For capacities in Btuh, divide by 12,000 to determine the total connected load in terms of nominal horsepower (or nominal "tons"). Terminal unit rated capacity at ARI Standard 320-93 performance conditions.

   Example: Unit rated 52,000 Btuh = 4.33 hp (tons)

2. Select evaporative cooler from manufacturer's data which indicate performance in terms of total capacity vs. summer design wet bulb temperature.
3. Apply diversity, if other than 80%, to permit selection of proper size cooler. Note that the flow rate remains constant for a given total connected capacity and wet bulb condition at any diversity. Never select cooler for 100% diversity (100% of the units running 100% of the time) or cooler selection will be oversized to no benefit. A few large units will require a larger diversity factor than a system composed of small increments. Probable diversity factors based on total system flow rate are:

   85% for up to 100 total system gpm (6.31 total system L/s)

   80% for 100 to 150 total system gpm (6.31 to 9.46 total system L/s)

   75% for over 150 total system gpm (9.46 total system L/s)

4. An alternate method of cooler selection for use with manufacturer's cooler performance curves can be found in "Miscellaneous Design Considerations," Chapter 7. Alternate methods of heat rejection are discussed in Chapter 3-A.

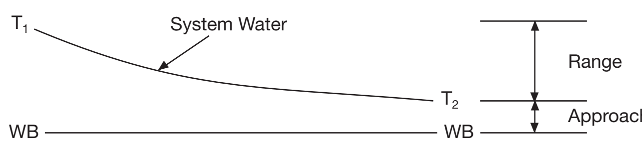
F. Determine loop water flow

The recommended loop water flow rate appears in the following table.

Outside Design W.B. °F (°C)	Temperature Leaving The Cooler °F (°C)	Flow Rate GPM/Ton (L/s/kW)	Cooler Range @ 75% Diversity °F (°C)	Approach °F (°C)
65 (18.3)	90.0 (32.2)	2.00 (0.036)	11.3 (6.3)	25.0 (13.9)
66 (18.9)	90.0 (32.2)	2.00 (0.036)	11.3 (6.3)	24.0 (13.3)
67 (19.4)	90.0 (32.2)	2.00 (0.036)	11.3 (6.3)	23.0 (12.8)
68 (20.0)	90.0 (32.2)	2.00 (0.036)	11.3 (6.3)	22.0 (12.2)
69 (20.6)	90.0 (32.2)	2.00 (0.036)	11.3 (6.3)	21.0 (11.7)
70 (21.1)	90.0 (32.2)	2.00 (0.036)	11.3 (6.3)	20.0 (11.1)
71 (21.7)	90.0 (32.2)	2.00 (0.036)	11.3 (6.3)	19.0 (10.6)
72 (22.2)	90.0 (32.2)	2.00 (0.036)	11.3 (6.3)	18.0 (10.0)
73 (22.8)	90.0 (32.2)	2.00 (0.036)	11.3 (6.3)	17.0 (9.4)
74 (23.3)	90.0 (32.2)	2.04 (0.037)	11.3 (6.3)	17.0 (9.4)
75 (23.9)	91.0 (32.8)	2.19 (0.039)	10.6 (5.9)	16.0 (8.9)
76 (24.4)	91.5 (33.1)	2.27 (0.041)	10.2 (5.7)	15.5 (8.6)
77 (25.0)	92.0 (33.3)	2.36 (0.043)	9.8 (5.4)	15.0 (8.3)
78 (25.6)	92.5 (33.6)	2.45 (0.044)	9.5 (5.3)	14.5 (8.1)
79 (26.1)	93.0 (33.9)	2.55 (0.046)	9.1 (5.1)	14.0 (7.8)
80 (26.7)	93.5 (34.2)	2.66 (0.048)	8.7 (4.8)	13.5 (7.5)
81 (27.2)	94.0 (34.4)	2.78 (0.050)	8.3 (4.6)	13.0 (7.2)
82 (27.8)	94.5 (34.7)	2.91 (0.052)	8.0 (4.4)	12.5 (6.9)

The table above indicates performance characteristics used in selection and cataloging of closed circuit evaporative water coolers, in terms of design summer wet bulb temperature.

Knowing the design wet bulb temperature for your area, simply enter the table and read cooling range, approach, and flow rate per unit capacity.



Range, T_1 - T_2	= 9.5°F (5.3°C)
Approach, T_2 - WB	= 14.5°F (8.1°C)
T_2 = WB + approach	= 78 + 14.5 = 92.5°F
	= 25.6 + 8.1 = 33.6°C
T_1 = T_2 + range	= 92.5 + 9.5 = 102°F
	= 33.6 + 5.3 = 38.9°C
gpm/hp	= 2.45
L/s/kW	= 0.044

Thus, a 100 hp (350 kW) system would circulate 245 gpm (15.5 L/s), entering the cooler at 102°F (38.9°C), leaving at 92.5°F (33.6°C), all at 78°F (25.6°C) WB.

1. Divide the evaporative cooler range by the system diversity to determine the range at the individual terminal units.
2. DO NOT apply diversity to the system flow rate. System cooling diversity only affects the range (and possibly the selected size) of the evaporative water cooler. Thus a 100 hp (74.6 kW) system, at 70% diversity, would circulate 245 gpm (15.5 L/s), entering the cooler at 101.4°F (38.6°C), leaving at 91.5°F (33.1°C), at 78°F (25.6°C) WB.

3. A higher flow rate presents no advantage, for increased annual pumping expense will completely offset expected improved performance of the terminal units.
4. Lower than recommended flow rates pose considerable disadvantage, for increased annual cooler and boiler operating hours diminish the inherent energy conservation benefit achieved in utilizing the closed water loop as a combination heat source/heat sink.

G. Establish any supplemental resistance heat for air preheat

If any electric resistance heat will be used to preheat the ventilation air coming into the building, establish the capacity of these heaters since it affects the heat addition required to the loop water. Determine the heating capacity of this equipment and enter on the design worksheet.

H. Establish any supplemental resistance heat used to offset glass radiation

If any electric resistance heat will be used to offset glass radiation losses, establish the capacity of these heaters since it affects the total capacity of the heat addition required to the loop water. Determine the heating capacity of this equipment and enter on the design worksheet.

I. Select the supplementary water heater

Supplemental heat may be added to the loop water by fossil fuel boilers, electric water heaters, or by steam or hot water heat exchangers.

Solar collectors may also be used to great advantage, assuming the implementation of thermal storage provisions to offset the factors of inclement weather persistence. Lacking adequate thermal storage, the solar system must provide only an alternate energy source.

1. **Conventional system without night setback:** Size the heater to match 70% of the building loss, *plus* the heat loss through the evaporative water cooler (This loss varies according to the degree of "winterization" provided in the cooler installation; see Step N for details).

   Alternate conventional system without night setback:

   a) Calculate building block heating load (Step B).
   b) Determine amount of electric resistance heat used to preheat ventilation air (Step F).
   c) Determine maximum amount of electric resistance heat used to offset glass radiation losses such as baseboard heaters or draft barrier heaters (Step G).
   d) Determine maximum net amount of heat to be supplied to the building by the water source heat pump units [This is equal to a minus (b plus c)].
   e) Size the supplementary heater to 70% of item d.

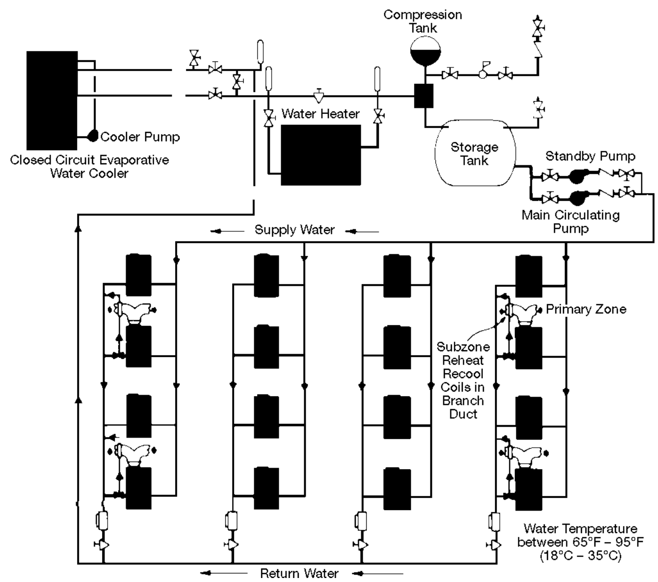
2. **Conventional system with night setback:** Size the heater to offset the heat of absorption of all units connected to the loop. As a rule of thumb, this is 8900 Btuh/ton (0.742 kW/kW) in terms of cooling load. If the morning startup includes simultaneous startup of cooling only process equipment, computer room units, and/or interior zone cooling only units, 80% of their heat of rejection can be safely counted on to reduce the heater size, but ensure that the minimum heater selected is equal to 70% of the building heat loss during the night setback period.

3. **Boilerless system:** Cold climate systems, where the water cooler (heat rejecter) is located outdoors, require a small instantaneous electric water heater to offset the heat loss through the cooler (This loss varies according to the degree of "winterization" provided for in the cooler installation; see Step N-9, page 20, for details).

J. Arrangement of major components in loop

It is typical to pump away from the electric water heater (or heat exchanger) as shown in the following diagram. However, it is also satisfactory to connect the water heater in the main from the heat pumps, just before the water cooler.

Water source heat pump system



K. Design of the closed system piping

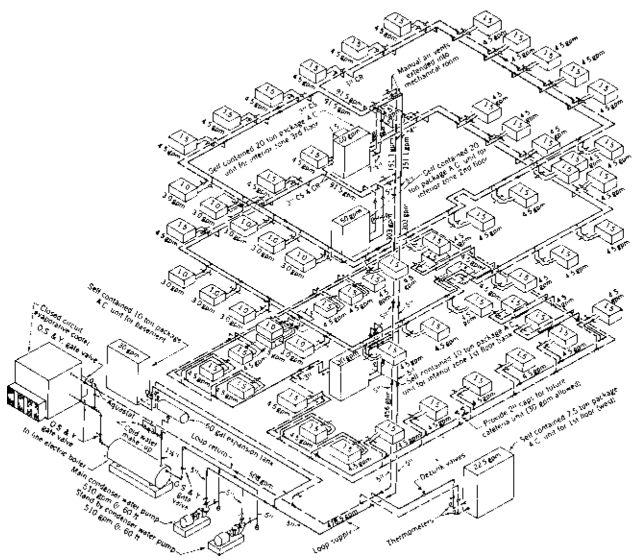
1. **Lay out piping:** Lay out piping to connect to all units. Use reverse return system whenever possible.

2. **Determine the flow rates:** Determine the flow rates in all sections of the system. This can be done only after establishing all flow rates (Step F). If unknown, determine the flow rate by the formula:

   Flow rate/unit = System flow rate/capacity x unit capacity

3. **Study the conceptual pipe circuiting arrangement:** To ensure that the pressure drop within each circuit and back to the pump are equivalent. Rearrange if necessary. Try to eliminate the need for balancing valves.

Isometric of closed loop system as designed for a particular building



4. **Size the pipe using the chart in this manual:** The chart is based on a maximum pressure drop of 4 feet per 100 feet (4 meters per 100m), and a maximum velocity of 10 feet per second (3m per second). For other conditions, use the accompanying graph.

5. **Calculate the friction loss in the piping:** Measure the length of the circuit to the least favored unit and back and multiply by 1.3 to allow for fittings to obtain the equivalent length of all fittings. Multiply the total equivalent length by the average pressure drop of 2.4 feet per 100 feet (2.4m per 100m) if using the chart.

6. **Calculate the total head on the circulating water pump:** Compile the different elements that make up the total head as follows:
   a) Friction loss in the piping.
   b) Friction loss in all heat exchange elements in the circuit such as the coil of the evaporative water cooler, boiler, etc.
   c) Friction loss in coil of least favored unit.
   d) Friction loss in all control valves in circuit (if used).

7. **Compute required pump power:** Since flow rate and total head are known at this point, power can be computed by assuming efficiency and using the following formula:

   Brake hp = (gpm x head x specific gravity)/ (3960 x pump efficiency)

   Water has a specific gravity of 1.0. With the addition of ethylene glycol for anti-freeze protection, the specific gravity increases as the percentage of glycol (by volume) increases:

   30% = 1.03
   40% = 1.05
   50% = 1.06

8. **Pump suction:** The pipe entering the pump suction should be straight for five pipe diameters, and the pipe should be the same size as the suction of the pump.

9. **Pipe material:** Generally use standard weight black steel pipe with black cast iron screwed fittings. For pipe sizes in excess of 2 inches (51 mm), it is common practice to use welded steel fittings. Where welded pipe is used, *Threadolets* or *Weldolets* for branches should be specified. Where the pressure in the piping will exceed 100 psig (689 kPa), use extra heavy pipe.

10. **Air vents:** Install manual air vents at the high points in the system to facilitate venting of air during initial fill. Air vents are not required on the individual terminal heat pump units, and should not be specified, as they unnecessarily increase the installed cost to no benefit. Air will be entrapped and carried to the system high points, unless the water flow rate is below the minimum flow rate for unit operation.

11. **Strainer:** The suction line of each pump should contain an installed cleanable strainer.

12. **Pipe supports and pipe expansion:** Ensure that all pipe lines include adequate pipe supports and provision for expansion. Provide required pipe anchors to accommodate expansion loops, joints, or bends.

13. **Valves:** System installation should include gate or ball valves as required to isolate equipment and piping zones for service. Install balancing valves in the system when it is impossible to design the same pressure drop in all circuits.

14. **Insulation:** Unnecessary on the loop water piping except on portions which run in unheated areas or outside the building, because loop temperature ranges between 65°F and 95°F (18.3°C and 35°C) and will neither "sweat" nor exhibit excessive heat loss.

Allowable flow rates for closed system piping

Standard weight steel pipe

Pipe Size (inches)	Flow Range (gpm)	Pressure Drop Range (ft/100 ft)	Maximum Total Connected Load①			
			2 gpm/10 MBH	2.2 gpm/10 MBH	2.4 gpm/10 MBH	2.5 gpm/10 MBH
0.5	0 - 2	0 - 4.00	10 MBH	9 MBH	8.5 MBH	8 MBH
0.75	3 - 4	2.5 - 4.00	20 MBH	18 MBH	17 MBH	16 MBH
1	5 - 7.5	2.0 - 4.00	27 MBH	34 MBH	33 MBH	30 MBH
1.25	8 - 16	1.25 - 4.00	80 MBH	73 MBH	69 MBH	64 MBH
1.5	17 - 24	2 - 4.00	120 MBH	110 MBH	105 MBH	95 MBH
2	25 - 48	1.25 - 4.00	240 MBH	220 MBH	210 MBH	190 MBH
2.5	49 - 77	2 - 4.00	385 MBH	350 MBH	335 MBH	310 MBH
3	78 - 140	1.5 - 4.00	700 MBH	635 MBH	610 MBH	560 MBH
4	141 - 280	1.25 - 4.00	1400 MBH	1270 MBH	1220 MBH	1120 MBH
5	281 - 500	1.5 - 4.00	2500 MBH	2270 MBH	2175 MBH	2000 MBH
6	501 - 800	1.75 - 4.00	4000 MBH	3635 MBH	3480 MBH	3200 MBH
8	801 - 1700	1.0 - 4.00	8500 MBH	7725 MBH	7390 MBH	6800 MBH
10	1701 - 2500	1.25 - 2.75	12500 MBH	11360 MBH	10870 MBH	10000 MBH
12	2501 - 3600	1.25 - 2.25	18000 MBH	16365 MBH	15650 MBH	14400 MBH
14	3601 - 4200	1.25 - 2.00	21000 MBH	19090 MBH	18260 MBH	16800 MBH
16	4201 - 5500	1.0 - 1.75	27500 MBH	25000 MBH	23900 MBH	22000 MBH
18	5501 - 7000	0.9 - 1.50	35000 MBH	31820 MBH	30435 MBH	28000 MBH

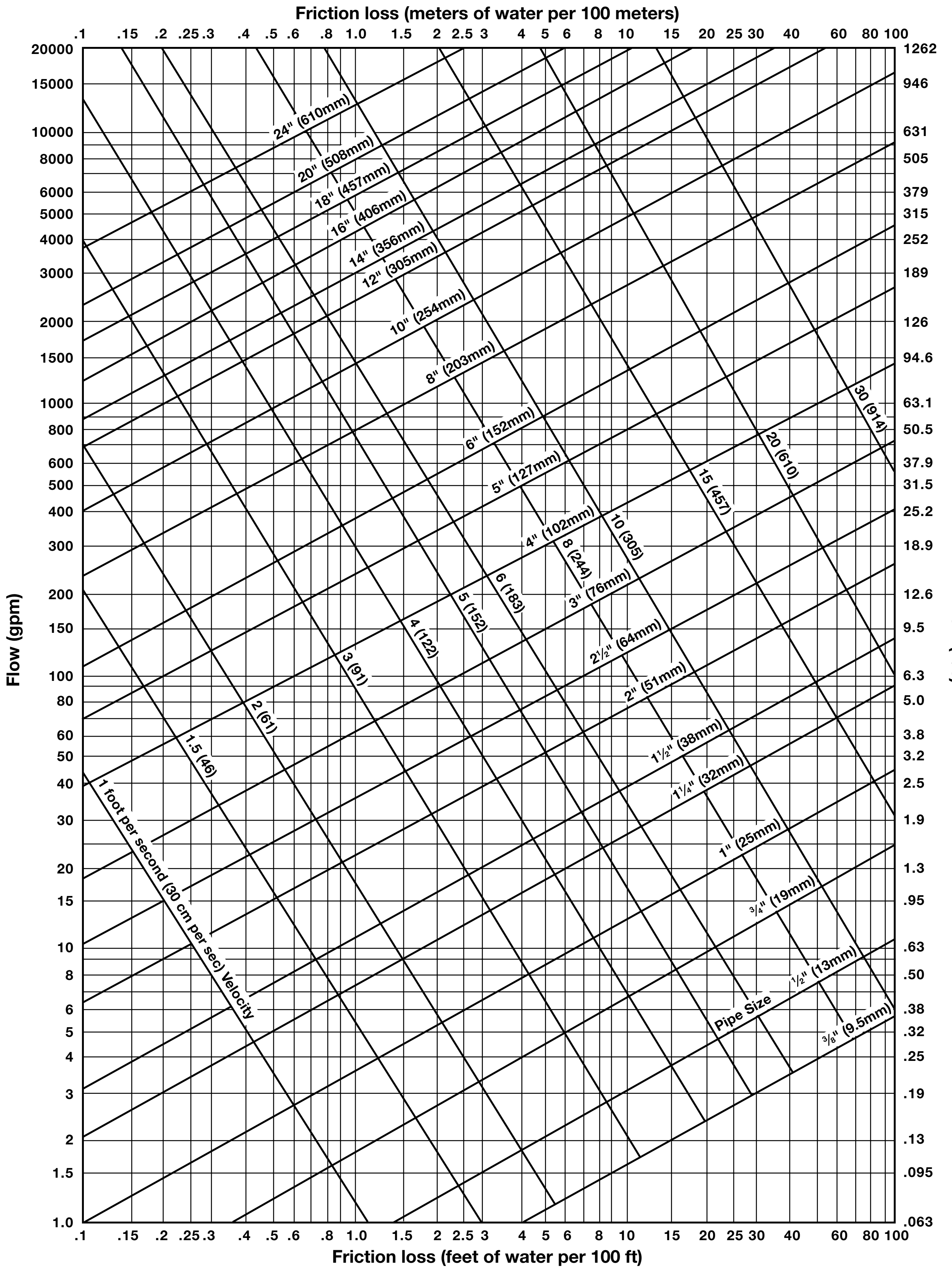
Note: The above capacities are based on a maximum pressure drop of 4 feet per 100 and a maximum velocity of 10 feet per second.
① Terminal unit cool capacity at ARI Standard 340-93 rating conditions.

Pipe Size (mm)	Flow Range (L/s)	Pressure Drop Range (m/100m)	Maximum Total Connected Load①			
			0.13 L/s/2.92 kW	0.14 L/s/2.92 kW	0.15 L/s/2.92 kW	0.16 L/s/2.92 kW
12.7	0.00 - 0.13	0 - 4.00	2.92 kW	2.63 kW	2.48 kW	2.34 kW
19.1	0.19 - 0.25	2.5 - 4.00	5.84 kW	5.26 kW	4.96 kW	4.67 kW
25.4	0.32 - 0.47	2.0 - 4.00	7.88 kW	9.93 kW	9.64 kW	8.76 kW
31.8	0.50 - 1.01	1.25 - 4.00	23.36 kW	21.32 kW	20.15 kW	18.69 kW
38.1	1.07 - 1.51	2 - 4.00	35.04 kW	32.12 kW	30.66 kW	27.74 kW
50.8	1.58 - 3.03	1.25 - 4.00	70.08 kW	64.24 kW	61.32 kW	55.48 kW
63.5	3.09 - 4.86	2 - 4.00	112.4 kW	102.2 kW	97.82 kW	90.52 kW
76.2	4.92 - 8.83	1.5 - 4.00	204.4 kW	185.4 kW	178.1 kW	163.5 kW
101.6	8.90 - 17.67	1.25 - 4.00	408.8 kW	370.8 kW	356.2 kW	327.0 kW
127.0	17.73 - 31.55	1.5 - 4.00	730.0 kW	662.8 kW	635.1 kW	584.0 kW
152.4	31.61 - 50.47	1.75 - 4.00	1168 kW	1061 kW	1016 kW	934.4 kW
203.2	50.54 - 107.3	1.0 - 4.00	2482 kW	2256 kW	2158 kW	1986 kW
254.0	107.3 - 157.7	1.25 - 2.75	3650 kW	3317 kW	3174 kW	2920 kW
304.8	157.8 - 227.1	1.25 - 2.25	5256 kW	4779 kW	4570 kW	4205 kW
355.6	227.2 - 265.0	1.25 - 2.00	6132 kW	5574 kW	5332 kW	4906 kW
406.4	265.0 - 347.0	1.0 - 1.75	8030 kW	7300 kW	6979 kW	6424 kW
457.2	347.1 - 441.6	0.9 - 1.50	10220 kW	9291 kW	8887 kW	8176 kW

Note: The above capacities are based on a maximum pressure drop of 0.4 bars per 100 meters and a maximum velocity of 3.048 meters per second.
① Terminal unit cool capacity at ARI Standard 340-93 rating conditions.

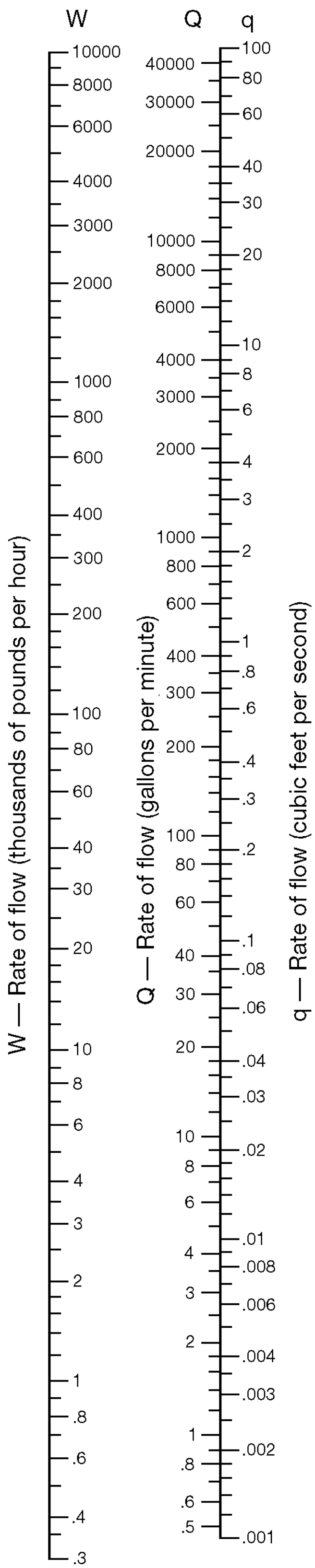
Flow graph for closed system piping

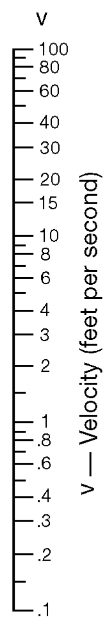
Standard weight steel pipe

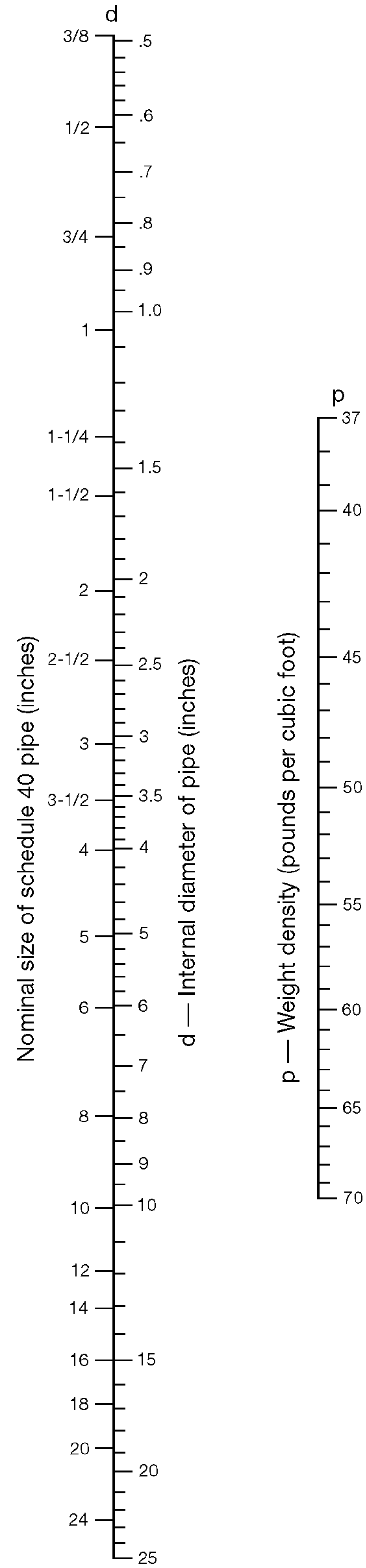


Example: For water flowing through a 2-inch standard (Schedule 40) pipe at the rate of 40,000 pounds per hour, find the rate of flow in gallons per minute and the mean velocity in the pipe.

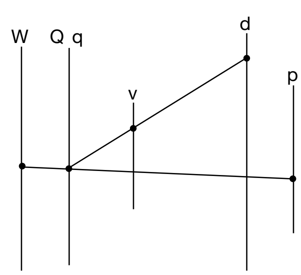
Solution: Assume the specific gravity of the water to be 1, corresponding to a weight density (p) of 62.4 pounds per cubic foot. Connect 40 on the "W" scale with 62.4 on the "p" scale; this line intersect the "Q" scale at 72 gallons per minute. Now connect 72 on the "Q" scale with 2.067 (I.D. in inches of 2-inch Schedule 40 steel pipe) on the "d" scale and note the intersection of this line with the "v" scale. The mean velocity of flow in the pipe is 6.9 feet per second.

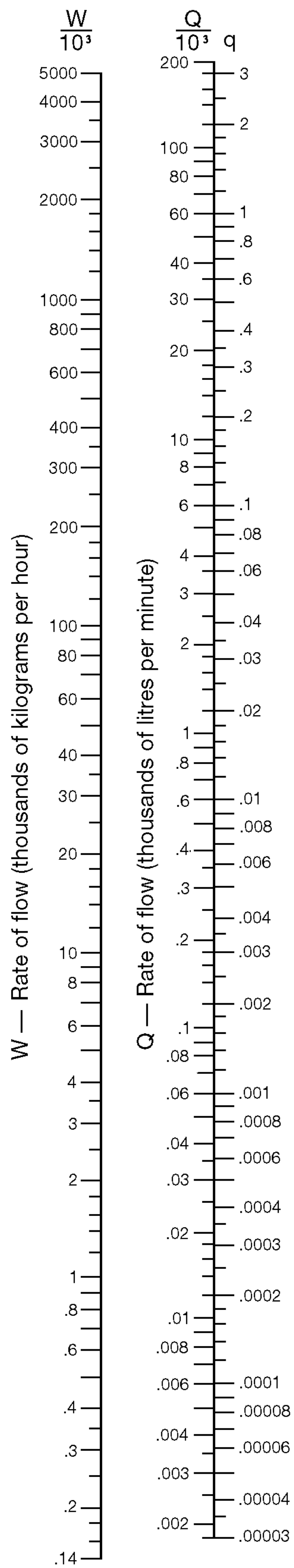






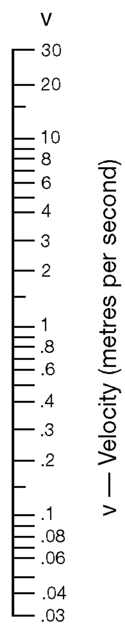
Courtesy of Crane Co.,
Stamford Connecticut.

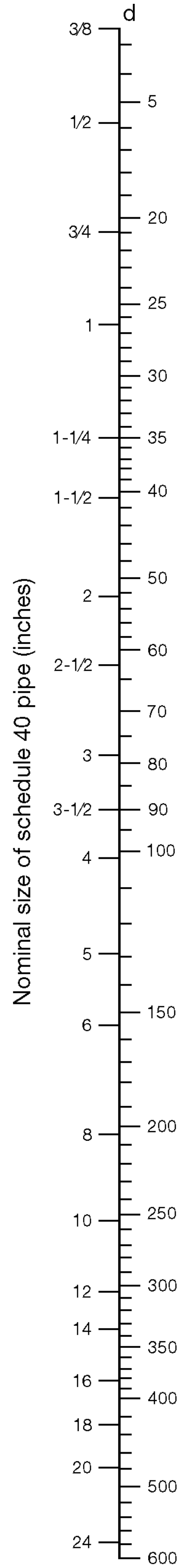


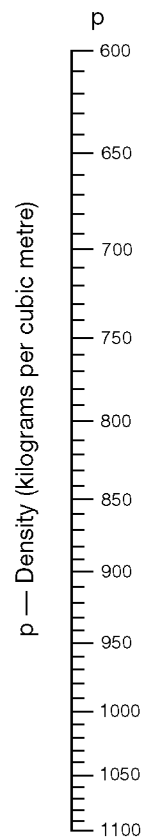


Example: For water flowing through a 2-inch standard (Schedule 40) pipe at the rate of 20,000 kilograms per hour, find the rate of flow in thousands of liters per minute and the mean velocity in the pipe.

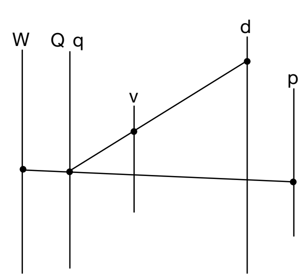
Solution: Assume the specific gravity of the water to be 1, corresponding to a weight density (p) of 1000 kilograms per cubic meter. Connect 20 on the "W" scale with 1000 on the "p" scale; this line intersect the "Q" scale at 340 liters per minute. Now connect .34 on the "Q" scale with 52.2 (I.D. in millimeters of 2-inch Schedule 40 steel pipe) on the "d" scale and note the intersection of this line with the "v" scale. The mean velocity of flow in the pipe is 2.6 meters per second.



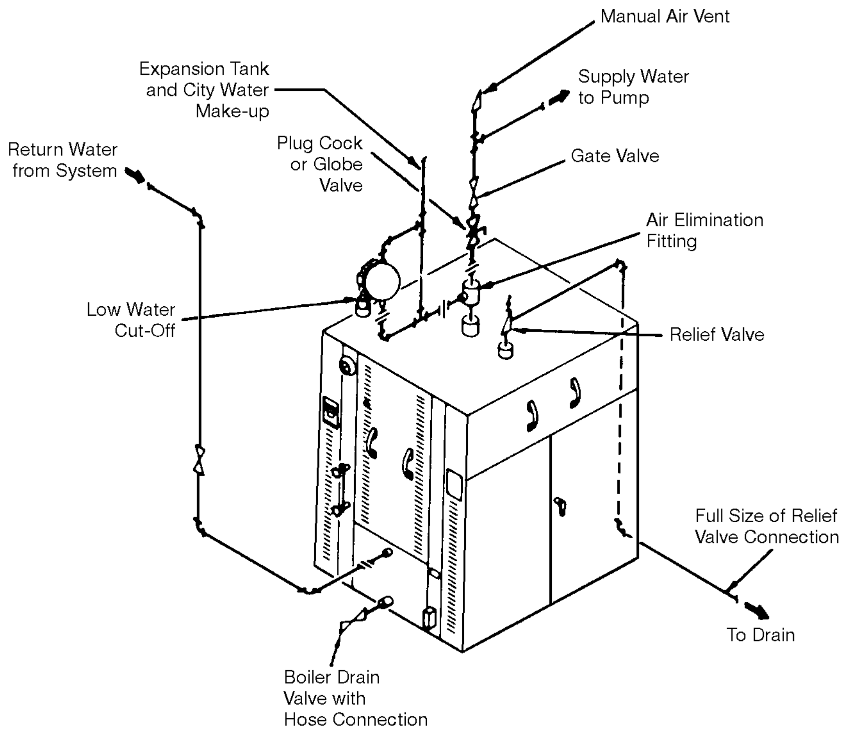




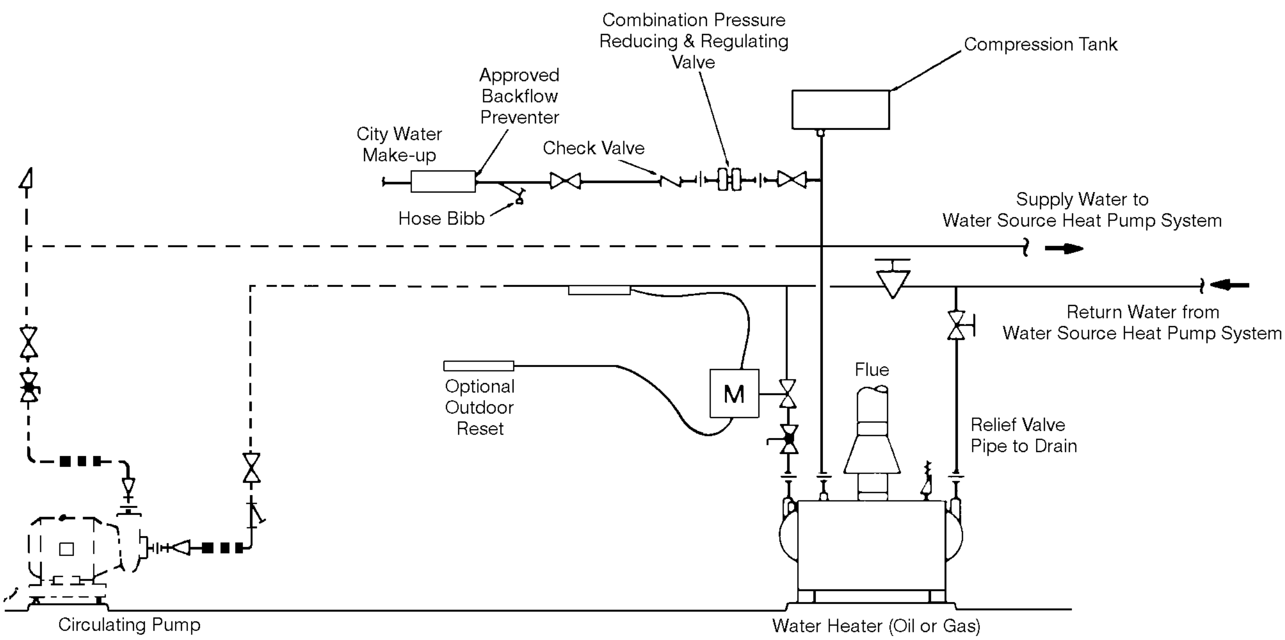
Courtesy of Crane Co.,
Stamford Connecticut.



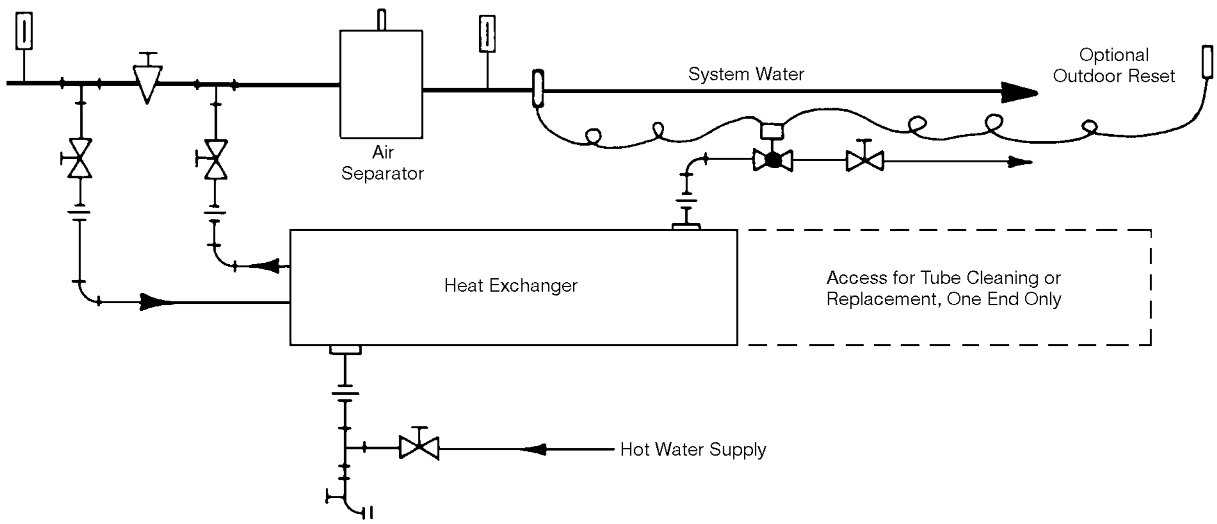
Piping details — electric water heaters

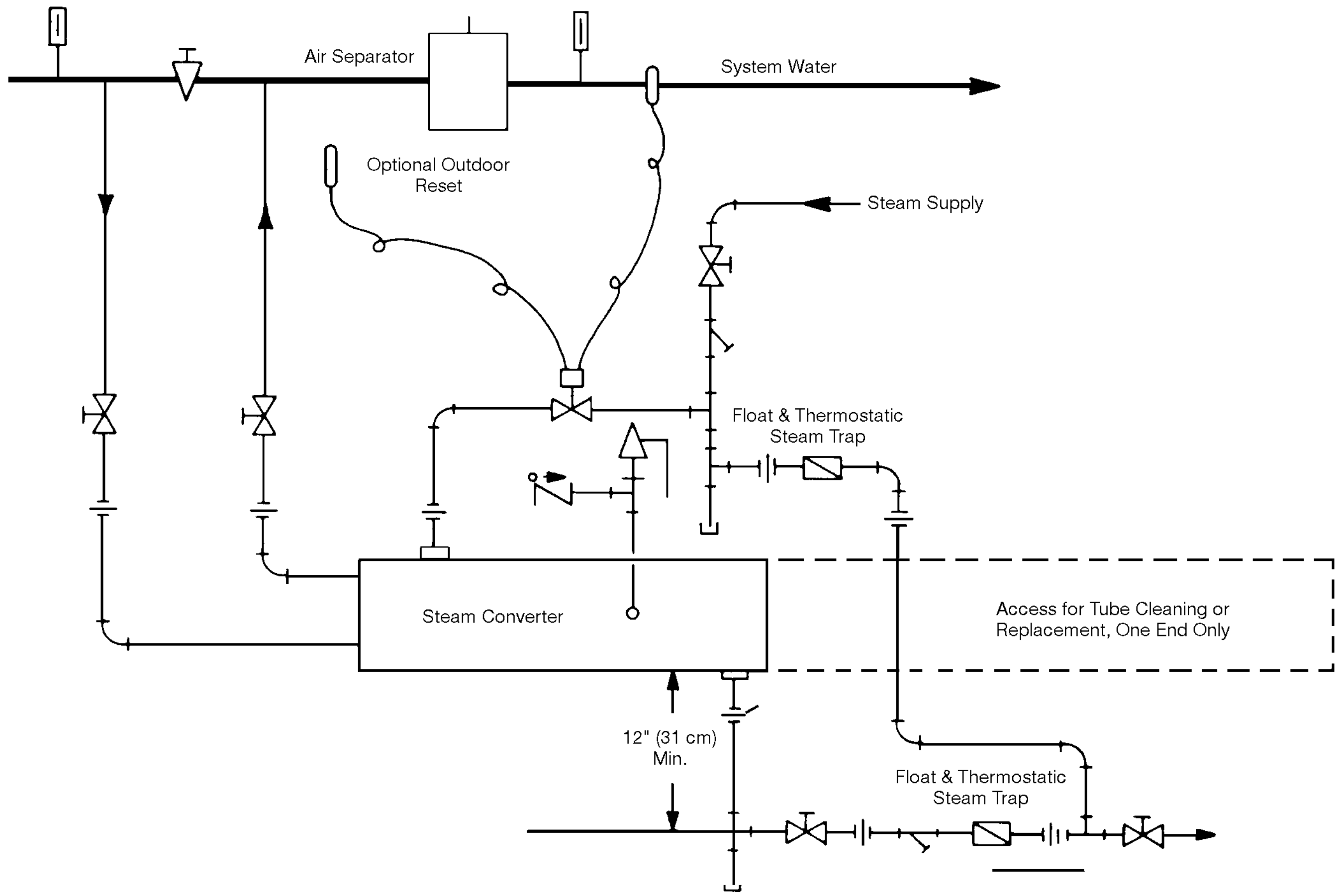


Piping details — gas or oil fired water heaters

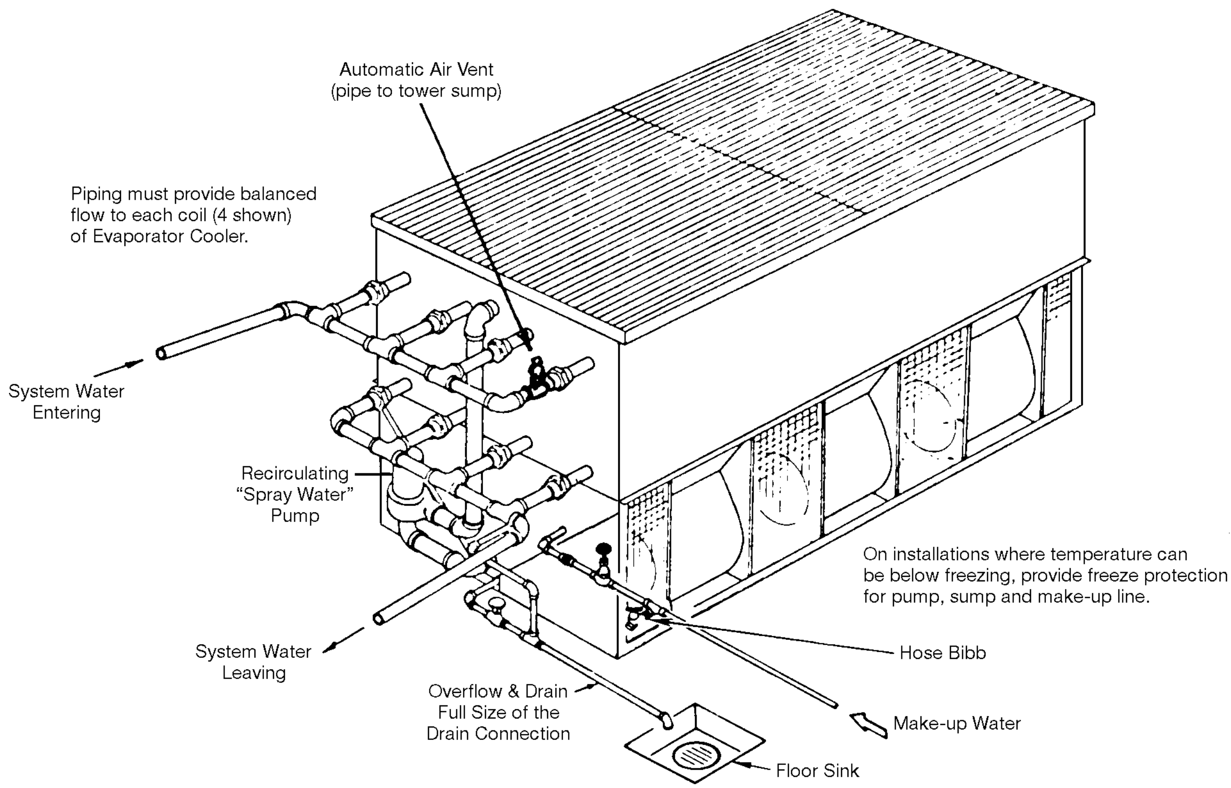


Piping details — supplementary heat
Heat exchanger — hot water or steam

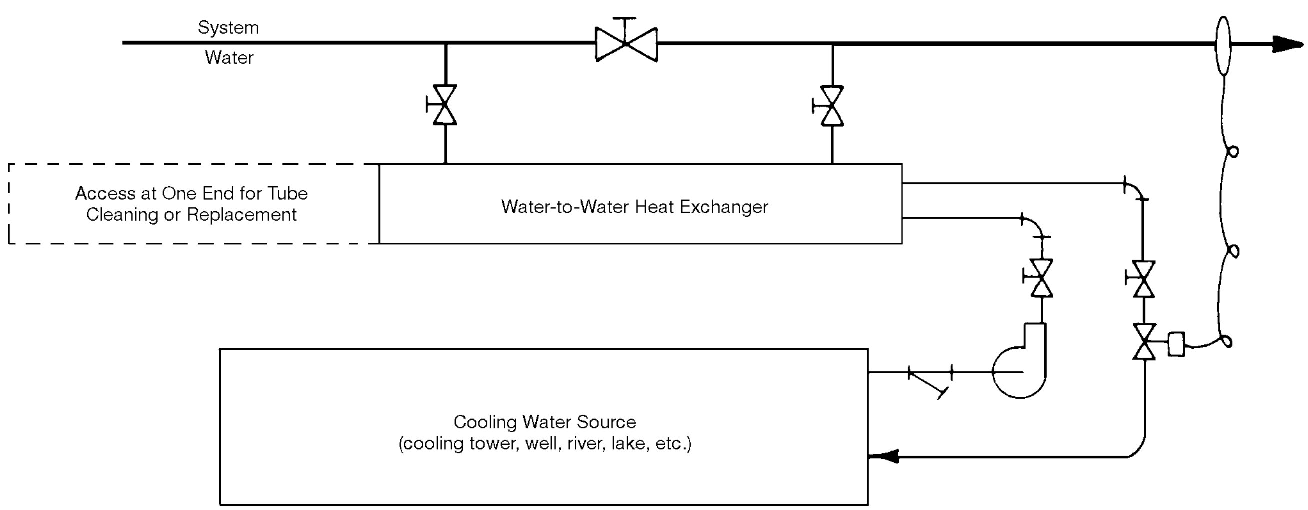


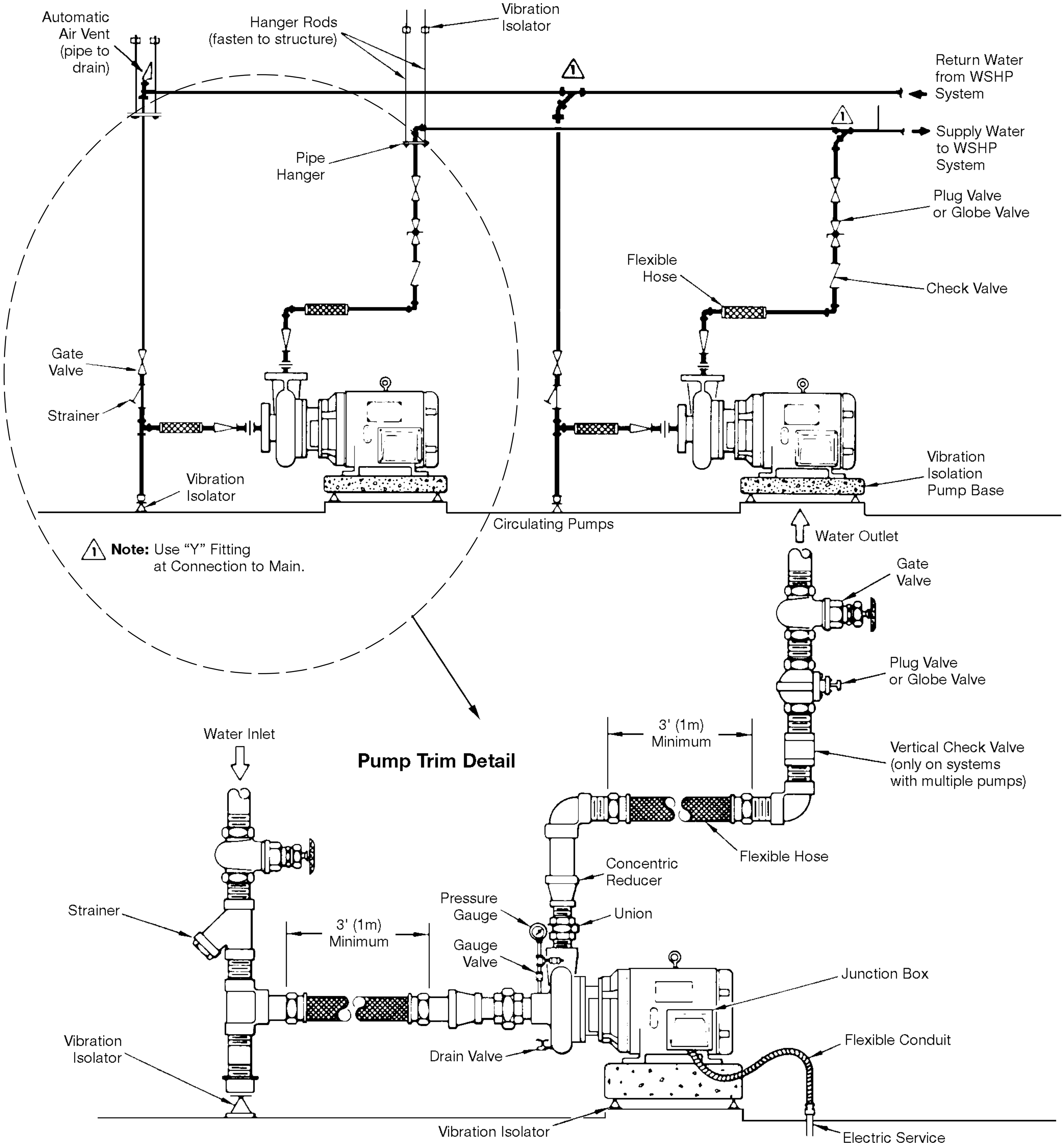


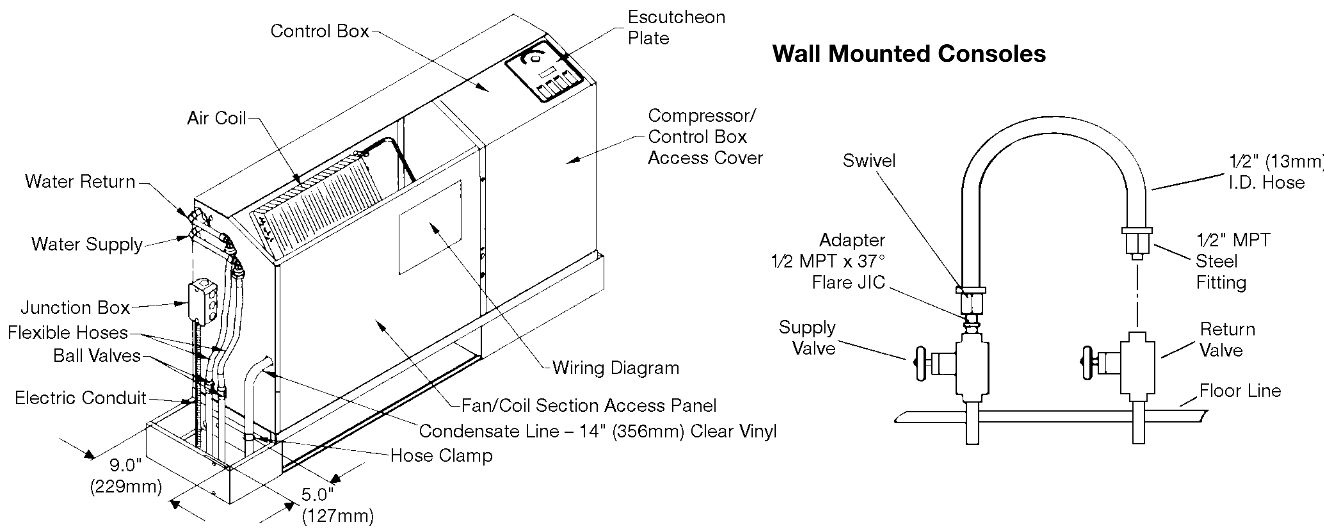
Piping details — evaporative water cooler



Piping details — heat rejection, shell & tube heat exchanger with open tower or well water

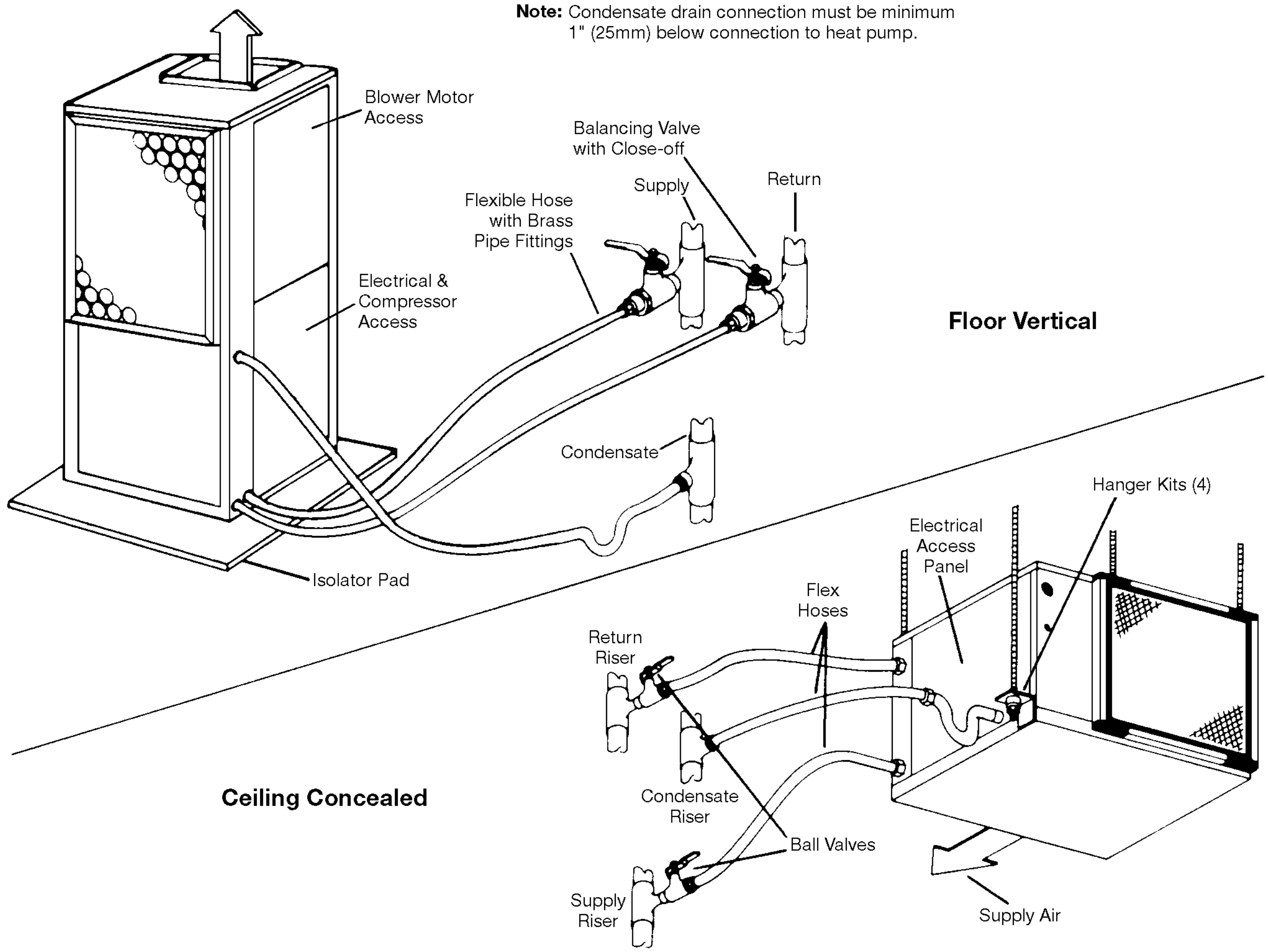



Automatic
Air Vent
(pipe to
drain)
Hanger Rods
(fasten to structure)
Vibration
Isolator
Return Water
from WSHP
System
Supply Water
to WSHP
System
Pipe
Hanger
Plug Valve
or Globe Valve
Flexible
Hose
Check Valve
Gate
Valve
Strainer
Vibration
Isolator
Vibration
Isolation
Pump Base
Circulating Pumps
Water Outlet
Note: Use "Y" Fitting
at Connection to Main.
Gate
Valve
Plug Valve
or Globe Valve
Water Inlet
Pump Trim Detail
3' (1m)
Minimum
Vertical Check Valve
(only on systems
with multiple pumps)
Flexible Hose
Concentric
Reducer
Strainer
Pressure
Gauge
Union
3' (1m)
Minimum
Gauge
Valve
Junction Box
Vibration
Isolator
Drain Valve
Flexible Conduit
Vibration Isolator
Electric Service



Wall Mounted Consoles

Note: Condensate drain connection must be minimum 1" (25mm) below connection to heat pump.



Floor Vertical

Ceiling Concealed

Note: All piping connections are field installed.

L. Pump selection

1. After pipe sizing and flow rate establishment, select the pumps. Flat head characteristic pumps are desirable, as a relatively constant head is necessary to ensure adequate flow to heat pumps at a distant point in the loop. This need arises due to potential short circuiting (excess flow) in low pressure drop machines connected closer in the loop.

2. It is considered normal good practice to specify a standby pump of equal capacity, piped in parallel, with check valves in each pump discharge.

3. An automatic pump sequencer is desirable since failure of loop water flow could cause freeze-up in the individual unit water coils in the heating mode.

4. The automatic pump sequencer should be considered mandatory if PVC piping is used.

M. Design of the cooling coil condensate drain piping

1. **General:** With air passage across the cooling coil, condensation occurs as the air reaches dewpoint temperature on the cold surface of the cooling coil. This moisture necessitates the provision of a drain pan under the coil, along with piping from the pan to a suitable termination point of disposal. The system of piping connecting all drain pans parallels any waste disposal system and must be designed to carry the water away without any significant maintenance.

2. **Stoppage and overflow:** In the absence of adequate design and installation of the condensate coil drain piping system, stoppage and overflow can result. When overflow occurs, considerable damage can result to building finishes. After building completion, the owner should be advised of the importance of a regular program of cleaning the condensate coil drain pan.

3. **Arrangement:** The condensate coil drain piping should reflect careful arrangement, allowing it to carry away the unwanted water. All parts of the system must be graded to drain, and wet portions or ungraded low points cannot be allowed since they will fill with dirt and stoppage and overflow will occur. Where horizontal runs are employed, piping should be pitched a minimum of 1" per 10 ft (8 cm per 10 meters).

4. **Calculation of water flow:** The amount of condensate that will occur in a system at the maximum condition can be computed from the design psychrometric chart for each unit. However, this entails a rather laborious procedure, especially if the system contains numerous terminal units. A rule of thumb of 3 lb/hr/ton (0.39 kg/hr/kW) may be followed if computing the water flow from each unit presents difficulty. Units serving areas with high latent loads may produce as much as 6 lb/hr/ton (0.78 kg/hr/kW).

5. **Available head:** Available head to cause the water to flow from the drain pan of the unit to the terminal point is the difference in elevation between the unit and the terminal point. Friction drop plays a negligible role in this calculation since the flow is always very small in relation to typical pipe sizes employed.

6. **Venting:** Venting of the cooling coil drain piping is more important than venting of the sanitary sewer system because the fan pressures can cause the water to hang up in the system. The potential pressure in a draw-thru unit could cause the air to come up the drain pipe and disrupt normal flow of all units in the system. All large units should have a vented trap of height 50% greater than the expected negative pressure in the pan. Locate this trap at the outlet of the drain pan.

7. **Material:** Generally, the condensate coil drain system construction should consist of PVC piping, thus eliminating the need for insulation (see Item 10). Alternately, if local codes prohibit the use of PVC, type “M” copper tubing should be used. When code interferes with the use of type “M” copper, type “L” copper or zinc-coated standard weight steel pipe should be employed. Copper tubing connections should consist of sweat fittings joined with 95-5 solder.

8. **Algae:** The development of algae in the pans and consequently down the drain system may occur in some geographical locations. When algae occurs, some form of chemical treatment may be necessary to keep the system open.

9. **Termination:** Several alternatives exist for the development of the termination point of the drop piping system. Generally, disposal of the accumulated water utilizing any system that complies with the local codes will be satisfactory. Spilling the water on grade usually proves unsatisfactory since it creates persistently muddy soil. Disposal of the water over a floor drain is also unsatisfactory since it keeps the floor surrounding the drain wet at all times.

10. **Insulation of the pipe:** The drop piping should be insulated with vapor barrier insulation since the contents can be quite cold and condensation may occur on the piping exterior, causing building damage. Insulating the pipe with 1/2 inch (13 mm) thick dual temperature glass fiber insulation or preformed flexible foamed rubber insulation prevents potential damage.

11. **Flushout:** The design of the cooling coil condensate drop system should allow periodic flushout to rid the system of sludge and dirt. The strategic placement of washout plugs will facilitate this procedure.

Sizing the cooling coil condensate drain piping

Pipe Size	Connected Cooling Load In Tons 50 100 150 200 250 300 350 400 450 500 550 600 650 700 750
¼"	Not Recommended
⅜"	Not Recommended
½"	Not Recommended
¾"	Up to 2 Tons Connected Cooling Load
1"	Up to 5 Tons Connected Cooling Load
1¼"	Up to 30 Tons Connected Cooling Load
1½"	Up to 50 Tons Connected Cooling Load
2"	Up to 170 Tons Connected Cooling Load
3"	Up to 300 Tons Connected Cooling Load
4"	Up to 430 Tons Connected Cooling Load
5"	Up to 700 Tons Connected Cooling Load

Note: Where horizontal runs are employed with a pitch of less than 1" per 10 ft. — increase the above values one pipe size.

Pipe Size	Connected Cooling Load In Kilowatts 175 350 525 700 875 1050 1225 1400 1575 1750 1925 2100 2275 2450 2625
6 cm	Not Recommended
10 cm	Not Recommended
13 cm	Not Recommended
19 cm	Up to 7 Kilowatts Connected Cooling Load
25 cm	Up to 17.5 Kilowatts Connected Cooling Load
32 cm	Up to 105 Kilowatts Connected Cooling Load
38 cm	Up to 175 Kilowatts Connected Cooling Load
51 cm	Up to 595 Kilowatts Connected Cooling Load
76 cm	Up to 1050 Kilowatts Connected Cooling Load
102 cm	Up to 1505 Kilowatts Connected Cooling Load
127 cm	Up to 2450 KW Connected Cooling Load

Note: Where horizontal runs are employed with a pitch of less than 8 cm per 10 meters — increase the above values one pipe size.

N. Design of the installation, closed circuit evaporative cooler (conventional system)

1. **General:** The closed circuit evaporative cooler differs from a conventional cooling tower in that the water to be cooled circulates through a closed coil inside the cooler, never experiencing atmospheric exposure. Systems accomplish evaporative cooling by pumping water from an open sump through sprays over the closed coil.

2. **Selection:** The heat rejection requirements of the system, taken from the completed design worksheet, should dictate the selection of the cooler.

3. **Capacity control:**

   a) **Modulating dampers** in the centrifugal fan discharge provide an accurate method of capacity control. A temperature sensing element controls the damper motor modulating the airflow through the tower. Constant water temperature maintenance at all load conditions provides excellent control for winter operation. A proportional reduction in fan motor power accompanies the reduction in flow.

   b) **Fan cycling** provides another method of capacity control. The temperature sensing element cycles the fan motors on and off. Control accuracy increases on multiple fan coolers.

   c) **Spray pump operation** should commence any time the outdoor temperature is above 32°F (0°C) and the air from the fans cannot provide enough capacity.

4. **Winter operation:** The vulnerability of the closed circuit evaporative cooler to freeze-up can lead to the problematic replacement of the expensive large steel coil inside the cooler. The following minimum steps should be observed:

   a) Provide a top outlet damper to close when fans stop.

   b) Insulate the entire casing and sump of the cooler with at least 2 inch (51mm) thick insulation.

   c) **Do not** modulate the water flow through the coil.

   d) Provide insulation and heat tracers on all exposed pipe including spray pumps and piping.

   e) Provide electric sump heaters or specify a heat exchange coil in the sump through which a small portion of loop water constantly flows.

5. **Spray water treatment:** Condensing system life relies on appropriate water treatment, which is determined by the condition of the air and water at the cooler location. Consult an experienced local company to obtain recommendations for proper water treatment.

6. **Bleed-off and make-up water:** Evaporative coolers evaporate approximately two gallons of water per hour per ton (2.2 liters of water per hour per kilowatt). With the replacement of only this amount, the concentration of impurities will soon have a harmful effect on the cooler. To prevent this, an additional two gallons per hour per ton (2.2 liters of water per kilowatt) should be bled off from the unit. The make-up water required is four gallons per hour per ton (4.4 liters per hour per kilowatt) or approximately 2.5% of the total water circulated.

7. **Location:** Location is a prime factor for consideration. Architectural compatibility and structural loading are obvious areas for coordination. Others, not so obvious, are:

   a) **Noise criteria:** Some cities have enacted noise codes, and specifications often require sound levels. Consult cooler manufacturers for octave band sound pressure ratings of the cooler and for assistance in sound evaluations.

   b) **Cooler fans** handle large quantities of air and their intakes and discharges should receive the same consideration as any other fan. Sufficient free and unobstructed space should exist around the unit to ensure adequate air supply. The possibility of air re-circulation, which reduces cooler capacity, should be carefully considered when installing the cooler near walls or in enclosures.

   c) **Avoid locations** near or down wind of stacks and incinerators to avoid the introduction of particulate matter into the cooler coils, which will load the coils and interfere with heat transfer.

   d) Try to avoid cooler locations which face the fan intake towards prevailing winter winds to minimize cooler heat losses.

8. **Piping:** When designing the condenser water supply and return piping to the cooler, exercise care to allow equal pressure drops when using multiple circuits through the cooler.

9. **Outdoor locations in cold climates** will produce a heat loss which demands consideration when sizing the supplementary heater. This loss through the cooler, with a 45 mph (20.1 m/s) wind, and a 60°F (33.3°C) temperature difference between outdoor air temperature and the water temperature is:

Degree of "Winterization" On Evaporative Cooler	Cooling Load kW/Ton① (kW/kW)①	Approx. Temp. Loss °F (°C)
a. Closure damper & factory insulation on damper & coil casing.	0.11 (0.031)	0.25 (0.14)
b. Closure damper only.	0.17 (0.049)	0.44 (0.24)
c. No damper — no insulation.	0.48 (0.14)	1.30 (0.72)

① Instantaneous net cooling load.

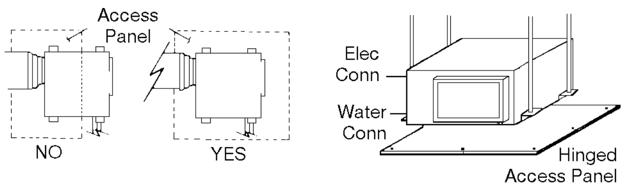
O. Design the location, access, ductwork and sound attenuation

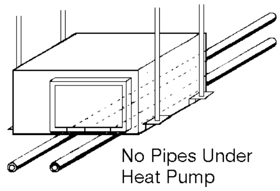
1. **Location of, and access to, heat pumps:** Providing the maximum accessibility for maintenance, service, or machine removal requires a coordination of trades. Remember:

   a) **All** mechanical apparatus require some maintenance.

   b) **All** mechanical apparatus will require service or replacement eventually.

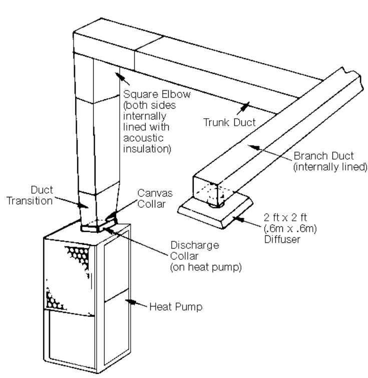
2. **Make a reflected ceiling plan** of lighting superimposed over mechanical layout.





3. **Specify ceiling access panels** under all ceiling mounted heat pumps, including:
   a) Clearance to hanger brackets, the two side panels, duct discharge collar, fittings and valves at water connections and electrical connections, both line and low voltage. Large hinged access panel or removable lay-in tile ceiling and T-bar is suggested. A minimum 18" (46 cm) clearance should be allowed on each side of the unit for service and maintenance access.
   b) Screwdriver access to electrical and blower service panels (two sides).
   c) Filter access for ceiling mounted with return air plenum. Leave slot for pulling filter straight down.
   d) Verify that the piping contractor does not run any lines directly under heat pump.

4. **Floor closet type:**
   a) Verify that the heat pump is mounted on a piece of rubber backed carpet a little larger than base area for isolation between machine and floor. Rubber backed carpet should be between ⅜ and ½ inches (10 and 13mm) thick (usually can be obtained as a remnant from a retail carpet store).
   b) Heat pump should be located with access to filter and service panel(s) at side(s) of machine. Consult manufacturer for location of access panel. A minimum 36" (91 cm) clearance should be allowed in front of each service access panel side, and a minimum 6" (15 cm) clearance should be allowed for filter access.
   c) Electrical conduit and pipe routing must not block filter removal. Filter often pulls straight up; sometimes it is removable from side. Also, conduit and pipe routing should not interfere with access panel.

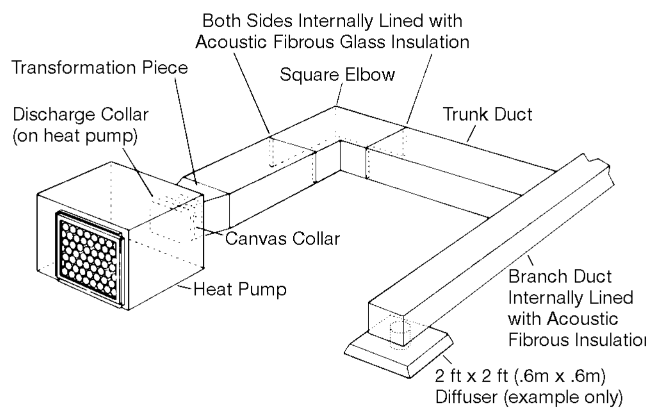


5. **Console type:**
   a) Some manufacturers furnish an enclosure that is installed first; some furnish an enclosure that may be lifted off for access to chassis.
   b) Coil covers over chassis supplied by manufacturer should not be removed until start-up. They perform the necessary function of keeping dust, trash and debris from falling into coils, condensate drain pan and fans, which could occur on a construction site.
   c) After enclosure installation, contractor should use empty carton as protective cover by taping onto installed enclosure and chassis.

6. **Large single-zone heat pump:**
   a) This model heat pump usually ranges from 10 to 30 tons (35 to 105 kW) and includes a single semi-hermetic compressor or multiple hermetic compressors. Fans are belt driven and machines weigh 1600 to 3300 lb (726 to 1497 kg).
   b) Rigging holes should be provided at ends of bottom channels. "Spreader" is recommended to keep cables clear of scraping upper portion of machine.
   c) Machine placement should allow a minimum 24" (61 cm) clearance for access to three sides (two ends and filter side) for removal of panels exposing electrical connections, piping pressure taps, blower and belt sheaves and compressor. A 6" (15 cm) minimum clearance at the back of the unit will allow for the removal of the screws holding the top panel. When allowing only minimum clearances on all sides of unit, top clearance is required for fan shaft removal.

7. **Ductwork and sound attenuation:**

Suggested duct layout for multiple diffuser application



   a) Ductwork is normally applied to ceiling, closet or floor mounted heat pumps on discharge side of machine. Such ductwork is relatively small (compared to central system ducts) so that it is often shop fabricated.
   b) A discharge collar is provided on all models to facilitate ductwork connection. The inclusion of a canvas connector is recommended between the discharge collar and duct transformation (enlargement). The preferred configuration for ceiling models, a horizontal transformation, typically requires a duct depth similar to the vertical dimension of the unit collar.

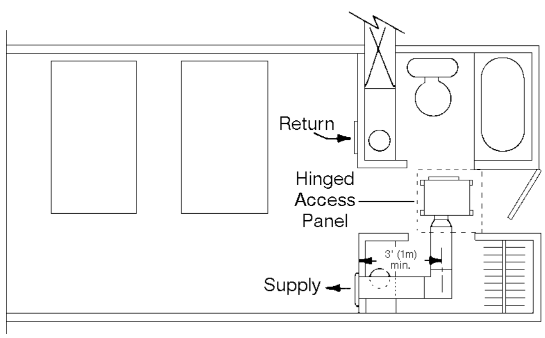
c) The heat pump location must allow the incorporation of a square elbow, without turning vanes, shortly after the transformation from discharge collar to full trunk duct to interrupt line-of-sight propagation of sound rays. One inch (25mm) acoustic fibrous glass duct lining should extend in both directions for a distance of at least two duct widths.

d) As a general recommendation, the interiors of ducts connected to heat pumps should be lined with acoustic fibrous glass of minimum ½ inch (13mm) thickness for full duct run. The only suggested exception occurs when the discharge trunk duct system feeds a series of air delivery light troffers.

e) For maximum attenuation, the last five diameters of duct before each air outlet (register) should be lined with one inch (25mm) fibrous glass blanket. Inside lining also serves as thermal insulation. Duct dimensions should allow for insulation thickness. See ASHRAE Guide.

f) Elbows, tees or dampers create turbulence and distortion in the airflow. A straight length of 5 to 10 times duct width is recommended to smooth out flow before the next fitting or terminal. Take-off of diffuser necks directly from the bottom of a trunk duct produces noise. If utilizing volume control dampers, locate them several duct widths upstream from air outlet. Check pressure drop for designed ductwork against external static pressure available with each machine at established airflow.

g) For a hotel, motel, dormitory or nursing home application, using a single duct register discharge from one machine, a velocity of 500 to 600 fpm (2.54 to 3.048 m/s) is suggested. These applications involve system static pressures as low as 0.05 inches of water (0.012 kPa) and duct lengths approximately six feet (1.8 meters). Discharge duct must include full lining and a square elbow without turning vanes.



Return air for these applications should enter through a low side wall filter-grille and route up the stud space to ceiling plenum. Return air ceiling grilles are not recommended.

h) For horizontal type heat pumps mounted in a hung ceiling, an attenuator box is sometimes placed at the air inlet to attenuate line-of-sight sound transmission through return openings.

i) For closet mounted heat pumps with return air through louvered doors, avoid line-of-sight connection between rear of louvers and air inlet to heat pump for maximum sound attenuation. Louver section should be boxed in and lined with one inch (25mm) acoustic fibrous glass if louver space does not permit a break in line-of-sight transmission.

j) Duct discharge from ceiling and floor mounted heat pumps usually enters conditioned area through:

Ceiling diffusers
Slotted type ceiling spline
High side wall registers or
One side of a heat recovery light troffer

P. Design the ventilation and exhaust system:

1. A wide variety of methods have proven successful at providing ventilation in buildings with water loop heat pump systems. The use of heat recovery units such as heat wheels and heat pipes is recommended.

2. Discharging the exhaust into the cooler, where the exhaust system design allows, will improve evaporative water cooler performance. The lower wet bulb temperature will improve summer operative efficiency, and the relatively warm exhaust will minimize winter heat losses.

3. In high-rise office buildings, normal practice dictates the introduction of ventilation air through interior zone equipment. Ventilation air should be introduced at each floor into a mechanical room, and there mixed with return air drawn back from the ceiling plenum. The mechanical room becomes a mixing plenum and usually contains a large capacity (up to 25 tons or 87.5 kilowatts) single-zone heat pump or a similar sized water-cooled packaged cooling unit with reheaters. Certain building configurations necessitate the utilization of more than one mechanical room per floor, requiring the installation of multiple heat pumps.

4. In either case, ventilation air often represents approximately 25% of total air supply and the outdoor air duct is usually equipped with a preheater so that the mixture entering the machine does not fall below 60°F (15.6°C). Air intake at outside wall normally includes a motorized damper interlocked with blower of machine.

5. For a low-rise building, ventilation air enters through one or more rooftop water source heat pump units connected to the loop and ducted down three or four floors to each ceiling plenum. The direct expansion unit is a standard coil depth and performs a tempering function on 100% outdoor air. Distribution of tempered air around a 24 to 30 inch (610 to 762mm) ceiling plenum encourages complete mixing with return air. A rooftop unit, equipped with duct heater, provides minimum mixture temperature of 60°F (15.6°C) in ceiling plenum. Outdoor air/return air mixture then passes through ceiling or floor mounted heat pumps which perform balance of sensible and latent cooling or heating requirements before delivery to each zone.

6. There are building configurations with load characteristics which permit a supply fan to introduce ventilation air uniformly into ceiling plenum(s) without tempering. Systems secure minimum inlet mixture temperature by bringing on ventilation fan(s) at a given interval after lights have been turned on. Use caution when locating return air near lights, thus adding heat, which adds to unit load.

7. Air often enters through one side of a heat recovery light troffer, slotted ceiling splines or lay-in diffusers arranged uniformly over full ceiling area (perimeter and core). This arrangement assures constant and controlled ventilation, free from wind pressure and stack effects, and makes

the incorporation of various types of effective outdoor air filters at one central point possible. Simplified filter maintenance procedures often result in minimized maintenance costs.

8. Manufacturers generally discourage the use of wall apertures at perimeter console heat pump terminals.
   a) Experience suggests a highly variable (and sometimes negative) amount of ventilation due to wind pressure changes on different exposures and to stack effect.
   b) Connection match-up at wall is often unsatisfactory and results in cold air leakage.
   c) The possibility of blow-through into room exists when using the less costly manual control type.
   d) Filtering is limited to fibrous glass or mesh media.
   e) This configuration drastically limits the effective filtering of polluted outdoor air, resulting in more frequent filter changes than required for recirculated air.

9. Positive exhaust blowers from toilet rooms, conference areas, and from return ceiling grilles at perimeter wall normally provide building exhaust. Some builders elect to duct exhaust air to the inlet of the heat rejecter where economy resulting from a reduced air temperature entering rejecter warrants the added cost of such ducting.

10. It is normal practice to interlock ventilation blowers and/or outdoor air damper motors, exhaust blowers and interior zone machines with time clocks. Clocks may be programmed to cut off all this equipment automatically during unoccupied periods (nights, weekends) in the case of cyclical operation such as office buildings, conference areas, restaurants, etc. This shutdown provides a simple and functional means to effect economies associated with "night setback." Heat loss, limited to transmission and infiltration, is offset by perimeter heat pumps operating on a continuous basis from their own thermostats.

Q. Design the temperature control system:

The factors of building usage, initial investment, and operating economy will determine the nature and complexity of the temperature control system. Night setback, ventilation control, and other simple control systems can dramatically reduce the system operating cost, and nearly always justify their expense. Much to the regret of several major control manufacturers, the water source heat pump system requires no large, expensive, or complex control packages to maximize its advantages.

In general, most installations will require that:

1. **Each individual air conditioner** forming a part of this system should include an overriding control arrangement which, in conjunction with one or more centrally located programming clocks, will accomplish the following:
   a) Restart all air conditioners after a general shutdown, from a central point, when so desired.
   b) Stop all air conditioners from the same central point when desired.
   c) Restart of the air conditioners as in (a) above must occur in random sequence to limit instantaneous current demand to a reasonable minimum.
   d) Keep all electric circuits to all air conditioners energized at all times to maintain a minimum conditioned space temperature 8°F (4.4°C) lower than the selected heat setpoint.
   e) Switch to cycled fan operation during the night shutdown period for areas with inherent or occupant selected continuous fan operation.
   f) Permit manual, timed override of stopped air conditioners, to enable occupants working at other than normal hours to restart their respective units for up to six hours of normal operation. Subsequent stoppage of the units at the conclusion of the timed override should not require a separate signal from the central control system.
   g) Permit **occupied**/**unoccupied** control of each zone from a central switch panel, which can override the signal to maintain continuous **occupied** (normal) or **unoccupied** (cooling off, and maintain space temperature as in (d) operational modes).

2. **Ventilation system** shutdown should correspond with individual heat pump shutdown, but initiation of restart should ensue only when the building occupancy or usage period commences.

3. **The timing of individual air conditioner restart** must include adequate provision for a morning "warm-up" period. This will vary depending upon outdoor temperature. With night setback to 8°F (4.4°C) below the normal heating setpoint, one hour usually satisfies both the restoration of space temperature and the removal of some "chill" from the furniture and walls.

4. Refer to Chapter 6 for a more thorough discussion of terminal unit control.

Chapter 2. Boilerless systems (all electric)

A. General description

The boilerless system features the elimination of the large boiler or supplemental heater in a closed loop water source heat pump system. Each reverse cycle air conditioner includes a resistance heating element and an automatic switch-over control, actuated by water temperature entering the unit.

When the amount of heat recovered and stored by the system falls below the amount of heat removed by units heating their zones, the loop water temperature will eventually drop to 65°F (18.3°C). When this occurs, the switch-over control in each individual unit will cycle the compressor *off* to discontinue heat pump heating, and a full heating capacity resistance heating coil cycles *on* to maintain the desired space temperature in that zone (see figures 1 and 2 below). Interior zone units in the cooling mode will continue to reject heat into the water loop system for storage. Eventually, the water loop temperature will rise to 75°F (23.9°C), at which time the individual perimeter zone units will switch back to the heat pump heating mode.

The design features an emergency override switch, so that the room occupant can override the automatic switch-over control, regardless of loop heater temperature, to provide space heating even in the event of compressor failure. This feature assures the occupant of heat at all times.

B. Schematic unit operational diagram (two modes of heating)

Figure 1.

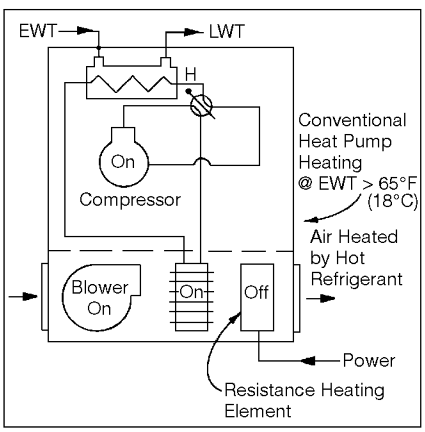
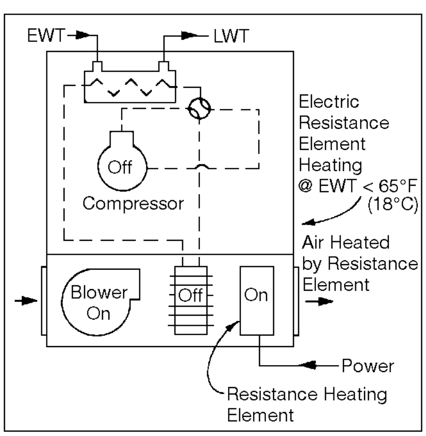


Figure 2.



C. Advantages

1. **Reduced first cost:** The installed cost of the individual resistance heaters with automatic switch-over controls is less than the installed cost of a central boiler. When available, factory installed resistance heaters provide an even more significant cost savings.

2. **Saves space** normally required for a boiler through the elimination of the boiler room.

3. Both owner and user benefit by **increased relability** of the decentralized heaters; neither a compressor failure nor a boiler failure will interrupt the heating.

4. System provides **better electrical diversity**, since individual energy conservation units switch from heat pump heating to resistance heating in random sequence as loop temperature drops.

5. **Compressor life is extended** through the reduction in annual operating hours.

6. The need for and **cost of a standby pump may be eliminated** since pump failure, a less critical predicament in the summer, does not interrupt the heating function.

7. **Energy conservation prevails**, as in a conventional water source heat pump system, since automatic switch-over from heat pump heating to resistance heating only occurs when the recovered heat stored by this system lags the amount required to heat the perimeter. The employment of the boilerless system induces no loss in the energy conservation inherent in the water source heat pump system.

8. **Smaller capacity equipment** can adequately handle high heat loss applications, thus further reducing first cost.

D. System "starter"

1. Systems located in cold climates, with an outdoor water cooler (heat rejecter), require a small instantaneous electric water heater to offset the heat loss through the cooler (This loss varies according to the degree of "winterization" provided for in the cooler installation).

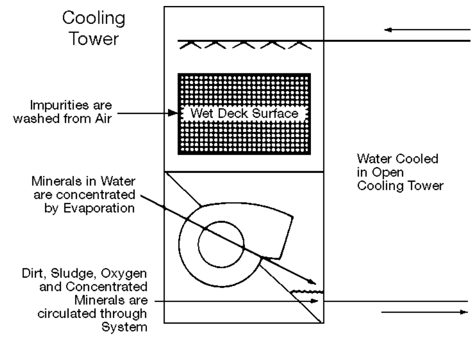
2. Size the "starter" in accordance with the degree of winterization provided on the evaporative cooler, as described in Chapter 1, Design Step N-9, page 20.

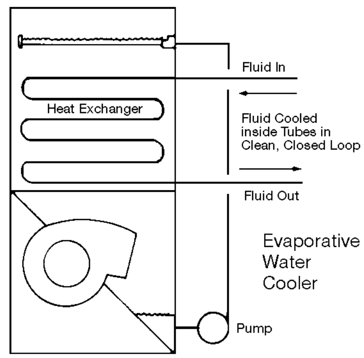
Chapter 3. System variations

A. Heat rejection variations

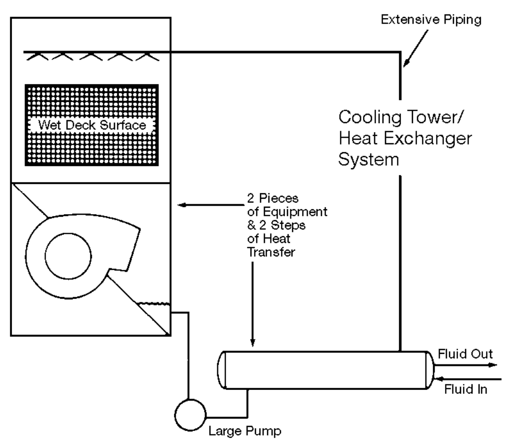
Where the closed loop system must reject surplus heat, it is general practice to employ an evaporative water cooler. Comparison with other heat rejection methods justifies the selection of this expensive piece of apparatus, as seen below:

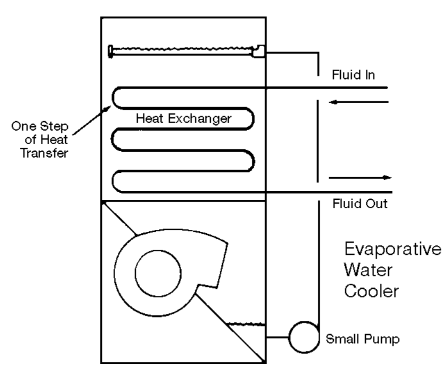
1. **Open cooling towers** operate on an evaporative cooling principle similar to evaporative water coolers, but differ in that the circulating water in the tower directly contacts the airstream. This method "washes out" impurities in the air, causing contamination of the cooling water. Open cooling towers reduce cooling water consumption 95% as compared to "once-through" systems, but do not eliminate the problems of scaling and corrosion. In fact, these problems escalate because the evaporation of the circulating water concentrates the impurities in the water. Also, the highly aerated water in an open cooling tower increases any tendency toward oxygen corrosion.
   The evaporative water cooler eliminates equipment fouling and scaling problems in process cooling systems by circulating the cooled fluid in a clean, closed loop system instead of an open system.





2. **Heat exchanger/cooling tower:** Combining two pieces of equipment — a cooling tower and a separate heat exchanger — can allow the achievement of the operating efficiencies of evaporative cooling and the maintenance advantages of closed loop cooling. Water from the cooling tower circulates in an open loop through one side of the heat exchanger and the process fluid from the cooled equipment circulates in a closed loop through the other side. The evaporative water cooler combines the cooling tower and heat exchanger in a single unit, thereby providing several distinct engineering and economic advantages.
   A single unit accomplishing two steps of heat transfer permits the realization of lower fluid temperatures.
   The evaporative water cooler requires a much lower water flow than a cooling tower/heat exchanger system of equivalent capacity. Consequently, the smaller required pump results in decreased pumping costs.
   The single unit evaporative water cooler generally translates to lower total installation costs than those of the tower and heat exchanger.





3. **Dry air coolers** can provide closed loop cooling but do not take advantage of the energy-saving evaporative cooling principle. The performance of dry air coolers, which usually consist of a finned-tube heat exchanger and several fans, depends on sensible heat transfer while the performance of evaporative water coolers depends on more efficient latent heat transfer.

   Because of the evaporative cooling principle, an evaporative water cooler can cool a fluid to within a few degrees of the ambient wet bulb temperature. In an air-cooled system, it may be practical to cool to within 15°F to 20°F (8.3°C to 11.1°C) of the ambient dry bulb temperature. Since design wet bulb temperatures generally lag design dry bulb temperatures by 15°F to 20°F (8.3°C to 11.1°C), evaporative cooling provides the opportunity to realize as much as 35°F (19.4°C) greater cooling.

   Utilizing the evaporative water cooler involves several other advantages over air-cooled equipment utilization:

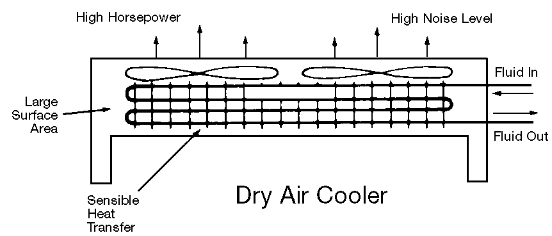
   a) Less heat exchanger surface required to cool equal loads generally leads to lower total investment cost.

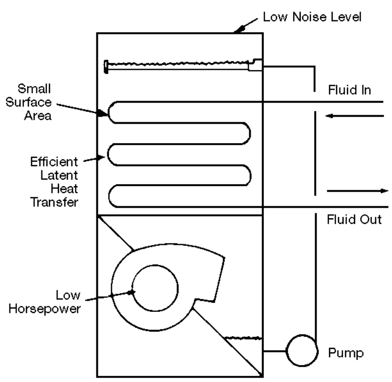
   b) The lower necessary air volume translates to lower fan power consumption. The lower airflow also translates to reduced noise levels, a definite benefit in areas enforcing strict sound standards.

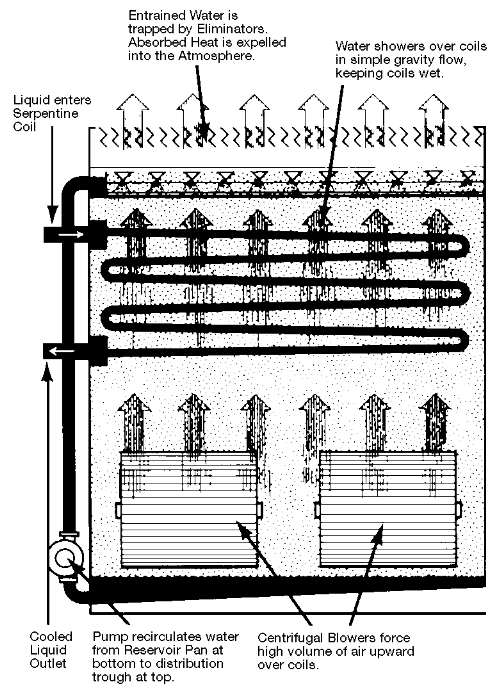
   c) Evaporative water coolers require less space and the compact design permits greater flexibility in location.

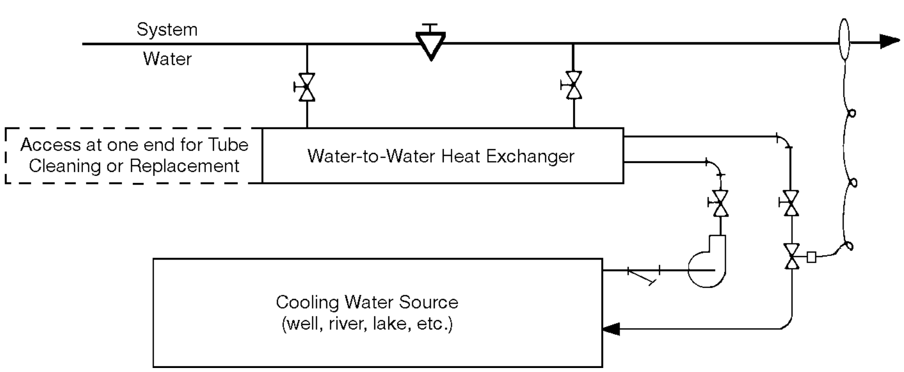
   d) When a convenient available water source exists, provided sufficient water quantity for adequate heat rejection, contractors may suggest a ground water heat exchanger. Using a water-to-water heat exchanger allows potential reduction of both initial installation and operating expenses.

   Typical coolant sources include wells, rivers, lakes or oceans. Performance of quality and quantity analysis of the coolant water ensures the use of proper heat exchanger materials and the proper fouling factor in the sizing procedure.









B. Energy storage supplement for equipment

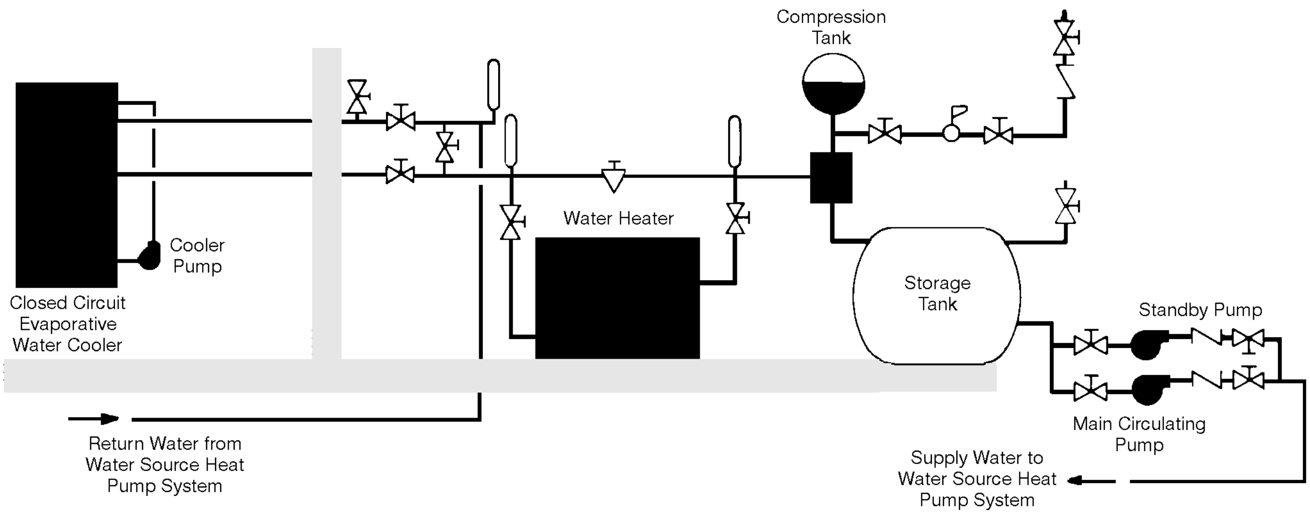
Increasing the mass of water in the closed water loop enhances the inherent energy conserving characteristics of a water source heat pump system. The additional mass, in a low temperature series tank, will act as a heat sink by absorbing any surplus energy generated within the building core. Conversely, the mass will act as a heat source for building night heating.

1. **Low temperature storage tank** — The low temperature storage tank will reduce the annual power requirement for both the supplementary water heater and the evaporative water cooler, along with leveling electrical demand.

Theoretically, a sufficient increase in the storage mass practically eliminates the need for a heat adder or a heat rejecter. Storage of the heat of rejection from summer operation could provide a heat source for the winter. This approach results in an excessively large required storage tank size, justifiably daunting prospective planners.

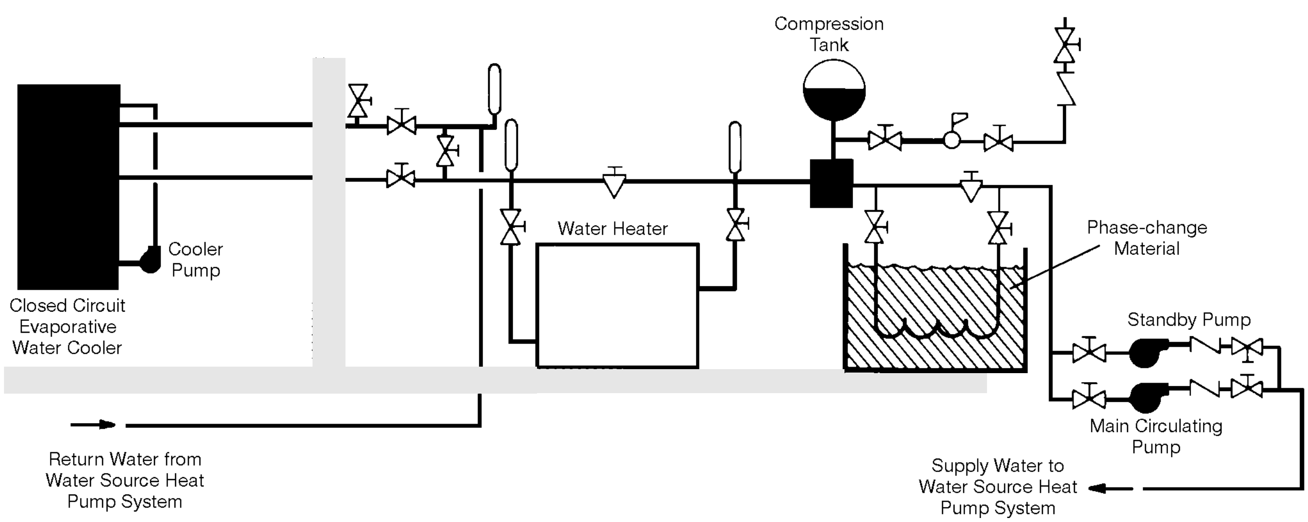
Practical sizing of the storage tank does not allow reduction in the size selection for the heat adder or the heat rejecter. Weather persistence will occasionally necessitate full capacity operation of these devices.

The incorporation of the water heater function into the storage both conserves space and economizes installation.



2. **Phase change storage tank** — The low temperature tank previously described utilizes sensible means to accomplish thermal storage, raising or lowering the temperature of the storage medium. Changing the physical state of the storage medium from solid to liquid or vice-versa can effect equivalent storage.

Heat storage through phase change provides a compact alternative because the heat of fusion of most materials greatly exceeds the specific heat. A phase-change storage tank for a water source heat pump system would be only one-fifth as large as a water tank of equivalent energy storage capacity.



A phase-change storage tank consists of an open, non-pressurized tank (95% fill) containing calcium chloride hexahydrate ($CaCl \bullet 6H_2O$), specially "doped" to limit supercooling (cooling below its freezing point without freezing). A copper loop heat exchanger immersed in this tank permits the transfer of heat from the loop water to the storage medium, or vice-versa. All, or a potion, of the loop water may pass through the heat exchanger.

The phase-change tank may also incorporate electric resistance elements for the purchase and storage of off-peak or low rate heating energy.

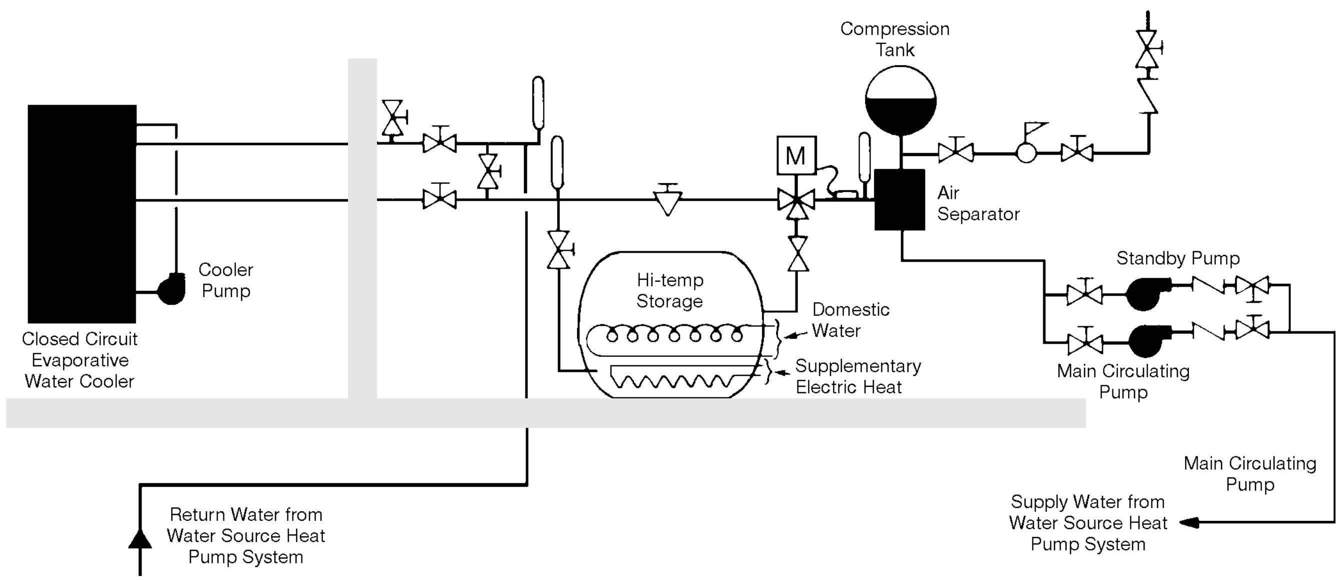
A comparison of water versus $CaCl \bullet 6H_2O$ characteristics illustrates the advantages of latent storage over sensible storage.

3. **High temperature storage** — The advantages of a high temperature storage tank appear under several conditions: high utility electrical demand rate, large heating load (over 5000 degree days), considerable demand for domestic hot water, electrical power time of day rate structures in use or anticipated, or certain combinations of these factors.

Electrical energy, purchased off-peak or during low rate periods, elevates the storage tank temperature to 180°F (82°C). The tank is piped in parallel with the closed water loop water as required to maintain the minimum loop temperature.

Commercial units permit storage at temperatures up to 280°F (138°C), and may conserve space and permit further economy of installation expense. However, a high temperature storage device will not absorb any excess heat generated in the building, and this will increase the annual evaporative cooler operating hours, when compared to low temperature storage methods.

A combination of high and low temperature storage methods may best suit certain projects.



4. **Storage sizing** — Generally, no "rules of thumb" for energy storage capacity exist. The selected storage unit size represents a compromise made in consideration of:

   a) Electrical energy costs (and anticipated future costs); in particular, the demand charge.

   b) Available space.

   c) The usage pattern of the building.

   Returning a system to normal operation following the night setback period has the greatest impact on electrical demand. During the building "warm-up" or "cool-down" periods, the terminal heat pump units will *all* operate, requiring that the water loop either supply their total heat of absorption (if heating), or absorb their total heat of rejection (if cooling).

   Typical water source heat pump systems have a water mass of 90 to 100 lb per nominal ton (11.7 to 13 kg per nominal kW) of installed equipment. The system components and the interconnecting piping contain this mass.

   If the water mass is at 90°F (32.2°C) and the system begins morning startup in the heating mode, depending on the length of the night setback (the magnitude of the structure's thermal inertia increases with the length of the NSB period), the outside ambient, assuming $1^1/_2$ hours of terminal unit operation to bring the building up to occupancy temperature; the energy stored in the basic water loop would be used up in 16 minutes (ΔT = 30°F = 16.7°C).*

   *30°F x 100 lb water = 3000 Btu storage

   *16.7°C x 45000 g water = 751500 cal = 3150 kJ storage

   (3000 Btu storage/11000 Btuh absorbed) x 60 min/hr =16.4 min

   (3150 kJ storage/3.2 kW absorbed) / 60 sec/min = 16.4 min

   For the remaining 74 minutes, the loop would require the addition of supplementary heat at a rate equal to the total heat of absorption of all the heat pumps. During this period, the system operates with a COP of 1.0. Full heater operation during this period increases both the power consumption and the demand rate, and the provision of a storage supplement can eliminate or minimize this eventuality.

   Once the building reaches occupancy temperature, simultaneous heating and cooling occur most of the time. In fact, throughout the winter, many modern buildings have a net cooling load during occupancy hours and a net heating load when unoccupied. A storage system permits the utilization of excess heat from daytime for night and morning startup heating.

   Due to the complex interrelationships between system components, a computer program provides the only practical analytical method to explore the exact effect of a particular storage mass on system operation. The designer must decide what benefits to expect from the storage supplement and make trial selections for computer evaluation and comparison.

   An allowable system configuration involves a combination of high and low temperature storage methods.

5. **Procedure** — To determine the optimum amount of stored energy:

   a) Determine the amount of heat (in Btu or kJ) required to raise the building from night setback to occupancy temperature at winter design conditions. Temperature with:

      1) Ventilation off.

      2) Night schedules for lighting, elevators, occupancy, equipment, etc.

   b) Divide this value by the sum total heating capacity in Btuh or kW of all the heat pump units.

   c) Multiply the quotient by 60 to determine the number of minutes required to bring the building up to occupancy temperature.

   d) Divide the sum total heat of absorption of all the heat pump units by 60, and multiply the quotient by the number of minutes run time from (c) above. The result is the amount of energy in Btu or kJ required from storage to prevent the water heater from being energized during morning startup.

   e) Determine the amount of heat required to maintain the building overnight and multiply by 0.65. The result in Btu or kJ is the amount of energy required from storage to prevent the water heater from being energized during overnight heating.

   f) Determine if the total energy of (d) and (e) is available as surplus from daytime system operation (or must be purchased): **heat to storage** from internal zone heat rejection, less heat transferred to exterior zones during daytime.

      Note 1: $A^1 Hr = Hr + \Delta H + A$

      For maximum storage, HR = 0

      Then, $A = A^1 Hr - \Delta H$

      And, $\Sigma A = TD\,(A^1 Hr - \Delta H)$ = heat transferred to tank

   g) If sufficient surplus heat *is* generated during daytime hours, determine tank size capable of absorbing and storing this energy.

   h) If the building has insufficient space available to accept a tank of the size as determined in (g), there are several alternatives:

      1) Provide storage for morning startup requirement only:

      $$\textit{Storage tank size}\ \text{(gal.)} = \frac{\text{Btu(d)} + \text{Btu(e)}}{8.34 \times \Delta T\ (°\text{F})}$$

      $$\textit{Storage tank size}\ \text{(liters)} = \frac{\text{kJ(d)} + \text{kJ(e)}}{4.15 \times \Delta T\ (°\text{C})}$$

      or

      2) Provide high temperature storage tank which, because of its higher ΔT, will be smaller in size:

      $$\textit{Storage tank size}\ \text{(gal.)} = \frac{\text{Btu(d)}}{8.34 \times \Delta T\ (°\text{F})^*}$$

      $$\textit{Storage tank size}\ \text{(liters)} = \frac{\text{kJ(d)}}{4.15 \times \Delta T\ (°\text{C})^*}$$

      *ΔT may be as high as 120°F (66.7°C) if the tank temperature is elevated at night with off-peak energy.

      or

      3) Do the best you can, since any increase in total system energy capability will reduce annual operating expense, and will have a favorable cost/benefit ratio (unless oversized beyond g).

Chapter 4. Water treatment

The cleaning, flushing and chemical treatment of a water source heat pump system is fundamental to efficient operation and the life expectancy of the system.

The following table demonstrates the major advantages of a closed loop heat pump system:

*Summary of water conditioning controls**

	Once-Thru	Open Recirculating	Closed Recirculating
Scale Control	1. Surface active agents such as polyphosphates. 2. Addition of acid. 3. pH adjustment. *Other considerations:* Adequate fouling factor Surface temperature Water temperature Clean system	1. Bleed-off. 2. Surface active agents such as polyphosphates. 3. Addition of acid. 4. pH adjustment. 5. Softening. *Other considerations:* Adequate fouling factor Surface temperature Water temperature Clean system	No control required.
Corrosion Control	1. Corrosion inhibitors in high concentrations (200 to 500 ppm). 2. Corrosion inhibitors in low concentrations (20 to 80 ppm). 3. pH control. 4. Proper materials of construction.	1. Corrosion inhibitors in high concentrations (200 to 500 ppm). 2. Corrosion inhibitors in low concentrations (20 to 80 ppm). 3. pH control. 4. Proper materials of construction.	Corrosion inhibitors in high concentrations. Proper materials of construction.
Slime & Algae Control	Chlorinated phenols. Other biocides. Chlorine by hypochlorites or by liquid chlorine.	Chlorinated phenols. Other biocides. Chlorine by hypochlorites or by liquid chlorine.	No control required.

*Abrasive materials must be kept out of the water system, and maximum velocity must not exceed those shown in Chapter 1.

The tremendous variety in water quality around the country makes the recommendation of a single best method of treatment impossible. Consult a local water treatment plant for specific treatment recommendations. This publication will address very general methods of water treatment for:

1. Closed loop water source heat pump system (closed recirculating).
2. Boiler with closed water loop, separated from water source heat pump system by heat exchanger.
3. Boiler (electrical) using same water as circulated through water source heat pump units.
4. Open recirculating system (closed circuit evaporative cooler sump).
5. Once-thru system (cooling only units).
6. Closed recirculating, separated from water source heat pump units by heat exchanger.

Water problems — Problems produced by the use of water fall into three general categories:

1. **Scale formation** — Mineral deposits which result from the crystallization and precipitation of dissolved salts in the water. The deposits form an insulating barrier, reducing the heat transfer rate and impeding the circulation of fluids due to increased pressure drop.
2. **Corrosion** — Decomposition of the metal caused by absorption of gases from the air. Corrosion may occur in any metal component of the system.
3. **Organic growths** — Slime and algae which form under certain environmental conditions, and can reduce the heat transfer rate by forming an insulating coating or can promote corrosion by pitting.

Water characteristics — The constituents of or impurities in water can be classified as dissolved solids, liquids, or gases, along with suspended materials. Filtration removes suspended materials but not dissolved materials. Determining potential problems requires an analysis of the water supply, together with the estimated system temperatures.

The characteristics of water important to our use are:

1. **pH value** — An arbitrary symbol used to express the degree of acidity or alkalinity. Neutral water has a pH of 7.0. Values above 7.0 to 14.0 are increasingly alkaline, while values below 7.0 approaching 0 are increasingly acidic. A pH below 7.0 promotes equipment corrosion. In water with a high pH (above 7.5 or 8), calcium carbonate scale deposits more readily.
2. **Alkalinity** — Sum of the carbonate, bicarbonate, and hydrate ions in water. Other ions such as phosphate or silicate may partially contribute to alkalinity. Generally, alkalinity defines the acid neutralizing power of the water. Determination of the scale forming tendency of water depends most heavily upon alkalinity.

3. **Hardness** — Sum of calcium and magnesium salts in water, although it may include aluminum, iron, manganese, strontium, or zinc. It is measured and expressed in parts per million (ppm). Carbonates and bicarbonates of calcium and/or magnesium contribute to the development of carbonate hardness (temporary). The remainder of the hardness, known as a non-carbonate (temporary) hardness, originates due to sulfates, chlorides, and/or nitrates of calcium and/or magnesium. Due to the fact that the solubility of non-carbonate hardness exceeds that of carbonate hardness by approximately 70 times, water conditioning to remove non-carbonate hardness poses less urgency for air conditioning systems.

Specific conductance — A measure of the ability of water to conduct an electric current, expressed in micromhos per cubic centimeter. The specific conductance indicates the tendency toward galvanic corrosion problems.

Water treatment — All water source heat pump systems and subsystems require water treatment. The type and degree of treatment requires appraisal of the numbers and types of water circuits, materials used in construction, temperatures, and water analysis. Each type of water circuit requires a different approach.

1. **Initial cleaning for all systems** — The initial cleaning and flushing is the single most important step. We recommend the procedures outlined in the ASHRAE Handbook, 1976 Systems, page 15.22.

2. **Closed recirculating systems** — These systems generally require no conditioning to prevent scale formation, and require no biocides for slime and algae control.

   Closed loop systems may require corrosion control. The treatment employed must protect against galvanic attack of any copper-steel couples. Various methods employed include:

   a) Sodium nitrate, borate, and organic inhibitors

   b) Sodium nitrate, borate, and silicate

   c) High chromate pH control

   d) pH and sulfite control

   e) Polyphosphate and silicates

   f) Alkalinity, phosphate, and sulfite control

   Because of the range in water quality encountered, a recommendation for or against any method poses difficulties, but selection of the inhibitors should include consideration of toxicity and the tendency of some inhibitors to stain (particularly chromates). Contact a local water treatment firm for an appropriate recommendation.

   The sodium nitrite inhibitor demonstrates compatibility with ethylene glycol solutions, occasionally used in northern climates or in solar loop subsystems.

   Consideration should be given to initial chemical charging of a closed system, to providing the owner with a practical method for supplemental treatment, and to testing and monitoring such a system.

   Even minor leaks of water at pumps or valve stems can require considerable makeup over a long period of time. Such makeup complicates the control of proper water treatment.

   Dielectric couplings serve no useful purpose in properly treated systems. The corrosion inhibitor, a required additive, makes them superfluous, providing no cost benefit to the user.

3. **Open recirculating system** — This system is not recommended for the water source heat pump units. Its continuous atmospheric exposure increases its tendency toward scale, corrosion, slime and algae formation.

   The performance and life expectancy of the heat pump units, in both heating and cooling modes, would suffer adverse effects if connected to an open recirculating system.

   The closed circuit evaporative cooler sump, by necessity, is an open recirculating system, thus requiring water treatment.

   Water evaporation from the outside of the heat exchanger may wash impurities from the air passing through the unit. The concentration of impurities increases rapidly and, if not controlled, can cause scaling, sludge, or corrosion, decreasing cooling efficiency or shortening equipment life. Bleeding some water from the pan can limit the impurity concentration. A bleed line, factory installed in the pump discharge, serves this purpose.

   With good water, the bleed rate may be as low as half the evaporation rate. Alternatively, a rate approximately equal to the evaporation may be required, with total water consumption ranging from a low of 2.4 gph per ton (2.60 liters per hour per kW), up to 3.6 gph per ton (3.89 liters per hour per kW).

   Extreme conditions may result in a bleed insufficient to control scaling and corrosion. A sound recommendation in this case mandates chemical water treatment of the pan water. Treatment selection must include the consideration of chemical compatibility with galvanized steel, along with maintenance of the pan water pH between 6.5 and 8.5.

   An automatic system to inject liquid conditioners into the basin provides the easiest method of providing scale and corrosion protection and micro-organism control. An alternate control method, the placement of acceptable briquettes in the basin in proportion to the flow rate, involves the periodic addition of replacement as required to maintain the proper concentrations. Selection of appropriate treatment requires consultation of a water treatment specialist familiar with local conditions.

4. **Once-thru system** — A once-thru system generally exclusively serves cooling only units. Serviced from city, lake, river, or well water supplies, it is not a distinct component of a water source heat pump system, although heat rejection frequently involves placing the closed water loop in heat exchange with a once-thru system. The product warranty varies with closed loop and once-thru systems; refer to the warranty document for details.

   A once-thru system may pose a scaling problem or a corrosion problem, typically not both. When requiring extensive water conditioning, economics may dictate the anticipation of a large scale factor and provision for frequent equipment cleaning and/or the use of corrosion resistant materials.

   Slime and algae, a frequent problem with lake and river waters, seldom pose any potential consequence with city or well water supplies.

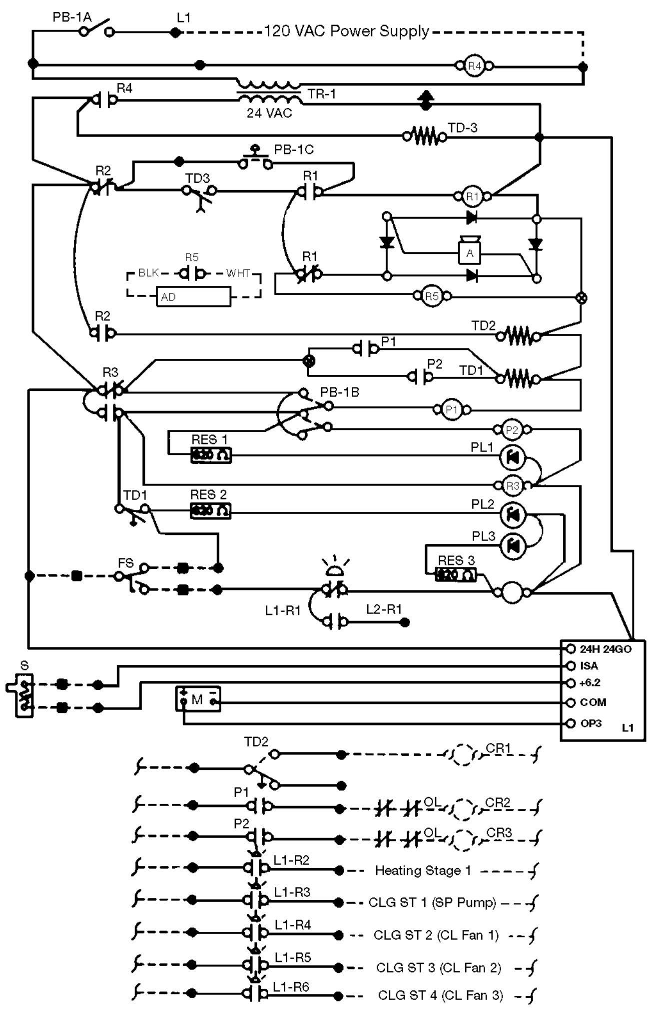
Chapter 5. Control of loop water temperatures

A. Control objectives

Control of the loop water temperature between a minimum of 65°F (18.3°C) in the winter and a maximum of 95°F (35°C) in the summer provides optimal system operation. Closer control may improve heat pump efficiency at the expense of reduced system efficiency. The controls should commence surplus loop water heat rejection in response to a temperature rise to 85°F (29.4°C) and achieve full capacity at 94°F (34.4°C). The controls should initiate supplementary heat addition to the loop when the loop water drops to 67°F (19.4°C). A potential advantage, realized through the utilization of automatic outdoor reset to raise the heating setpoint in the winter, requires calibration of all supplementary heating functions to de-energize at 80°F (26.7°C) on a temperature rise, and prevent re-energization above 77°F (25°C) on a temperature drop.

A solid-state system safety and operating panel will provide necessary control functions plus alarm functions for high and low temperature and loss of flow, indication of loop water temperature, and allowance for manual pump sequencing. Optional remote alarm indicators are available.

Typical SSOP control wiring diagram

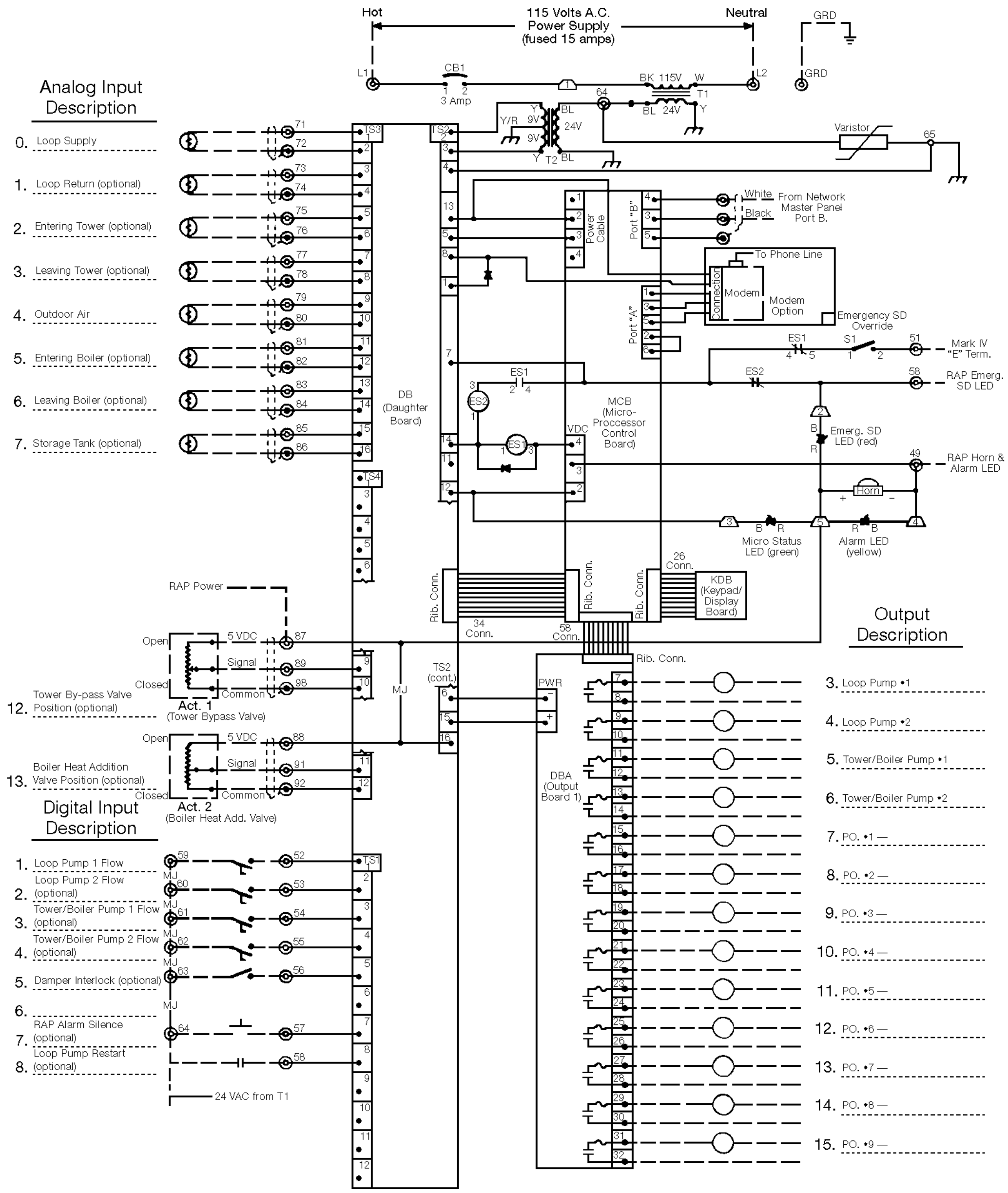


Legend:

Symbol	Description
|◀	Diode
■	Wire nut
⊗	Tap connection
●	Terminal block connection
(◀)	Light emitting diode
○	Component tie point
– – – – –	Optional wiring by others
———	Wired by McQuay International
--( )--	Field installed relay by others
A	Solid state alarm
AD	Auto dialer
CR-1	W.S.H.P. unit interlock
CR-2/3	Pump starter
FS	Flow switch
L1/L2 – R1-R6	Controller
M	Temperature meter
OL	Overload
PL1	Red light
PL2	Red light
PL3	Green light
P1	Pump relay no. 1
P2	Pump relay no. 2
PB-1A	On-off switch
PB-1B	Pump selector switch
PB-1C	Alarm silence switch
R1	Alarm silence relay
R2	Safety operating relay
R3	Standby pump relay
R4	Restart relay
R5	Optional alarm interface relay
RED-1/3	Resistors
S	Temperature sensor
TD-1	Pump changeover time delay relay
TD-2	System cutout on internal malfunction time delay relay
TD-3	Alarm delay time delay relay
TR-1	Transformer

Alternatively, McQuay International offers the MicroTech Loop Water Controller, a microprocessor-based control panel designed to provide sophisticated control and monitoring of the loop water temperatures. Primary control features include: outputs for heat rejection, heat addition, and time clock control, automatic or manual pump sequencing, tower loop and boiler pump control, occupied and unoccupied time scheduling, up to eight temperature readouts, alarm indicating emergency shutdown, precool and preheat cycles, manual control, and a communications port.

Typical loop water controller schematic



Legend:

Symbol	Decription
	Factory wire terminal
	Field wiring terminal
	Field wiring
	Printed circuit board terminal
	Cable — sheilded, twisted, and jacketed pair with drain wire
	Thermistor temperature sensor
	Current or voltage signal device

B. Circulating pump control

The control method selected for the main loop circulating pumps will depend upon several factors such as:

1. Type of pipe used (PVC pipe requires special consideration).

2. Night shutdown of pumps requires interlock to prohibit terminal unit operation during the off period.

3. Night shutdown of pumps requires consideration of any elements of the closed loop located outside the building, possibly subject to freezing.

Standard practice dictates the specification of a standby pump of equal capacity, piped in parallel, with check valves in each pump discharge. The normal desire to reduce first cost by eliminating the standby pump should be resisted, as the building cannot be cooled without loop water circulation, and cannot be heated without loop water circulation unless the terminal units are of the "boilerless" type. Water may freeze in the heat rejecter coil, a component very expensive to replace, unless the loop water contains antifreeze.

A prudent design recommendation includes the specification of an automatic pump sequencer capable of energizing the standby pump whenever the main pump fails to operate or provide the necessary flow rate. The sequencer unit should also incorporate a manual lead-lag selector switch, facilitating routine pump maintenance and/or repairs.

A differential pressure switch, installed across the pump's suction and discharge, is the preferred method of sensing flow failure. A paddle type flow switch also suffices if installation avoids proximity to elbow, where turbulence may "fool" the control.

PVC loop water piping offers many attractive benefits, which may dictate its use, but certain control arrangements should be considered mandatory. The use of PVC necessitates the installation of an automatic pump sequencer *or* some means of de-energizing the terminal units in the event of loss of flow. *Both* are desirable. When flow ceases, a terminal unit will continue to operate until activation of its internal safety controls. For operation in the cooling mode, it will trip the high pressure switch, generally at about 414.7 psia (2859 kPa) or 153°F (67°C). The water in the heat exchanger will also reach this temperature. If flow resumes before the water in each unit cools down, the exposure of PVC loop piping to temperatures violating design limits may cause the piping to sag or joints to come apart.

Night pump shutdown requires interlock with terminal unit controls, thus preventing routine cycling of terminal unit safety switches. Additionally, portions of the water loop may lose heat through outside walls, causing terminal unit exposure to entering water far below design low temperature limit on morning startup. A several minute time delay between pump startup and terminal unit startup will eliminate this eventuality.

Night pump shutdown also requires thermostat override during periods of low outside ambient to prevent freeze-up of portions of the loop outside the building, including the heat rejecter unit. For these reasons, the main circulating pump generally operates continuously. However, loop water pump power generally represents about 6 ½% of the total energy consumed by the system, and proper controls allow pump power savings of approximately 35%.

C. Heat rejection control

Surplus heat rejection control systems should incorporate some means of capacity control to minimize required purchased power. The following control steps generally lead to maximum energy conservation:

On a temperature rise:

Stage 1 — Open closure dampers (heat rejection by convection).

Stage 2 — Energize spray pumps (heat rejection by evaporative cooling).

Stage 3 — Energize fan motor (heat rejection by increased rate of evaporative cooling).

On larger coolers with multiple fan motors, motor staging will minimize power consumption. Some single fan motor coolers include a two-speed motor option.

Modulating dampers in the fan discharge, recommended when the cooler may operate at or near freezing temperatures, provide some conservation of fan power.

The heat pump manufacturer can best determine the temperatures corresponding to the initiation of various stages of heat rejection. The manufacturer weighs the increasing power consumption and decreasing capacity of the terminal units as loop water temperature rises against the power required to reject surplus heat and the operational limits of the terminal units. This selection must also provide sufficient differential to prevent short cycling of large fan motors.

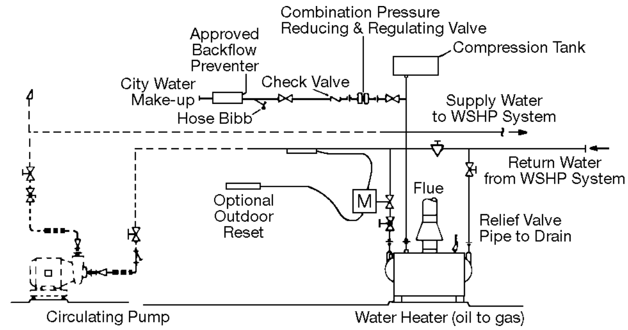
A properly designed control system will also incorporate time delays to limit current in-rush on startups following brief power interruptions, common during summer thunderstorms.

Some designers may wish to drain the cooler sump during periods of freezing temperatures, or skip the spray pump stage, to simulate dry cooler functioning. This option requires consideration of pan water heater sizing. Heaters generally provide only sufficient energy to prevent the pan water from freezing in an idle cooler. Preventing freezing in a cooler operating dry generally requires additional capacity. Therefore, a drain down step is required. Reduced air resistance and increased airflow during dry operation require oversized fan motors to prevent overloading.

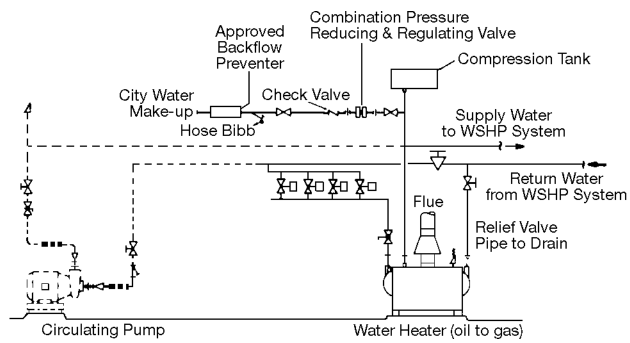
D. Supplementary heat control

When the temperature of the closed water loop approaches the lower limit, it becomes necessary to add heat to the water. The amount of heat required will vary from just enough to offset the difference between heat rejected by units operating in the cooling mode and heat absorbed by units operating in the heating mode, to full boiler capacity to offset the heat of absorption of all the units, as may occur during morning warm-up or during a winter design heating period. Modulation of heating capacity is therefore desirable.

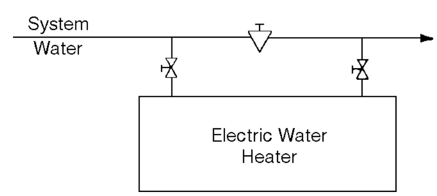
Fossil fuel heater — Oil or gas heaters require maintenance at relatively high temperatures to prevent flue gas condensation within the heater. For this reason, they are piped in parallel with the closed loop. A two-way modulating valve mixes high temperature heated water with returning loop water as required. The separate thermostat furnished with the heater allows control of the high temperature jacket.



Alternately, a series of inexpensive two-way motorized zone valves may be paralleled as shown below. A system safety and operating panel or loop water controller controls these valves for thermal staging. This approach often involves a relatively low total installed cost, and the multiple valves provide greater reliability than the single modulating valve.

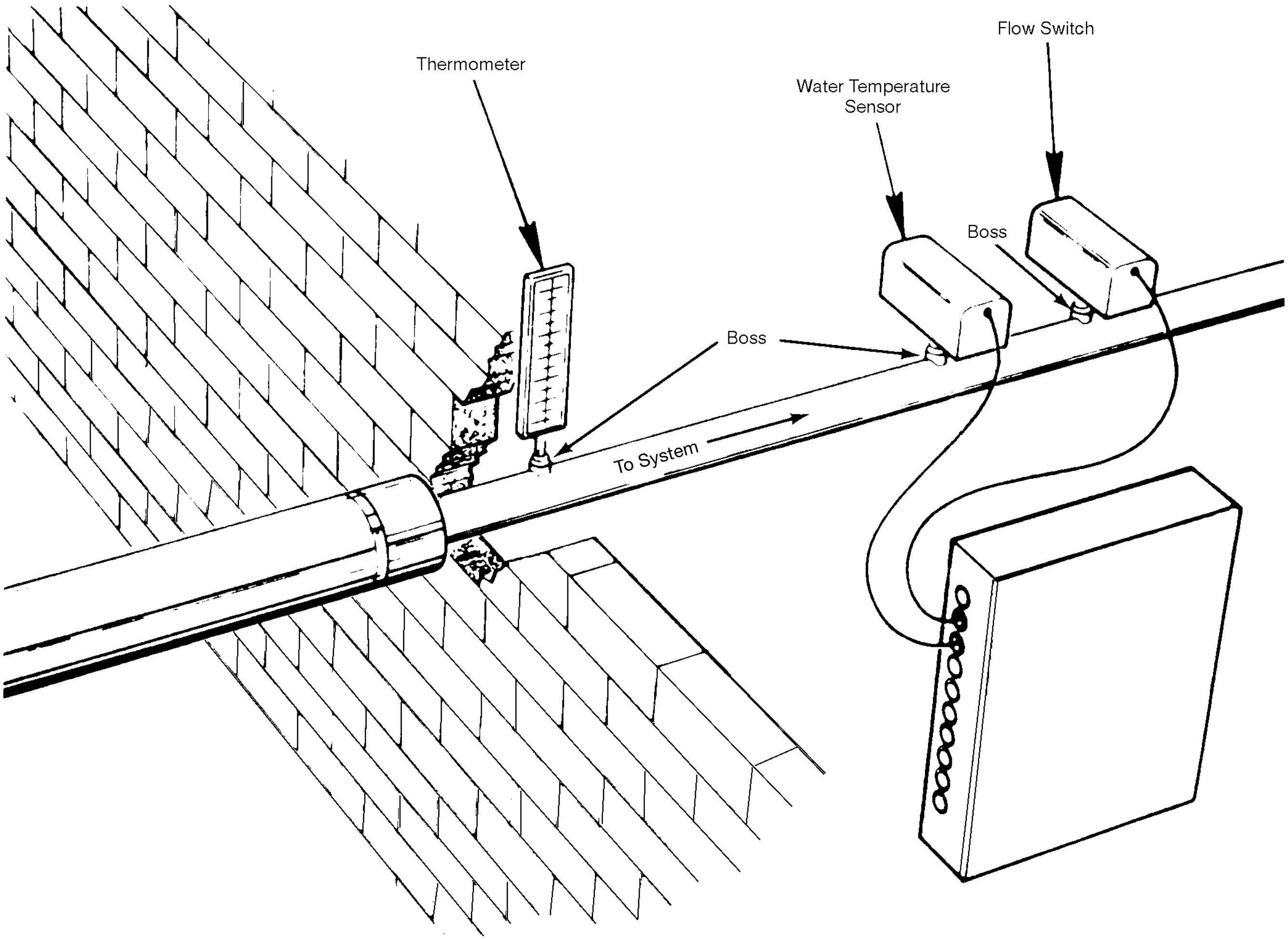


Electric water heater — An electric water heater may accept the circulation of the full flow of loop water, or alternately, only a portion as shown below. All but the smallest heaters will have a step controller to provide a time delay between the energization of the heating elements, limiting current in-rush. Steps or groups of steps may be staged. The number of circuits available in the heater, the capacity (load rating) of the switching device contacts, and the allowable temperature rise per step (a function of accuracy desired) will all contribute to the determination of the number of steps or stages.



Outdoor reset — A variable low limit to loop water temperature may be used to advantage. The terminal unit's heating capacity increases with warmer entering water, so an outdoor reset control can be used to raise the loop temperature during periods of low outside ambient; e.g. increase loop temperature to limit 15°F (8°C) as outside temperature drops to 0°F (18°C).

System control sensing point — Temperature sensing for all loop temperature controls should occur in the water line leaving the evaporative water cooler. Normal practice involves pumping away from the electric water heaters, as shown on the piping circuit diagrams. However, it is also satisfactory to connect the water heater in the main from the heat pumps, just before the water cooler. Regardless of the selection of major component arrangement, the control sensing point remains the same — in the line leaving the cooler.



E. System safety controls and alarms

System controls must contain limit devices to perform safety shutdown of certain functions, and should signal occurrence of the abnormal condition.

A high temperature fault, as might occur due to a broken fan belt in the evaporative cooler, should activate an audible alarm and cause shutdown of the terminal units. A normally closed contact, arranged to open in the event of a high temperature condition, may also serve as a backup safety in the supplementary heat control circuit. The high temperature limit is normally set around 105°F (41°C).

A low temperature fault, as might occur due to failure of the supplementary heater, should activate an audible alarm and cause shutdown of the terminal units. The low temperature limit is normally set around 57°F (13°C).

Loss of water flow, a critical contingency, will generate serious consequences during freezing temperatures. Many installations typically reject some heat during freezing temperatures where loss of system water flow could cause immediate freeze-up of the cooler coil. Additionally, the normal stepped control of the supplementary water heater, with time delays between de-energization of elements, may not respond fast enough to prevent heater damage. Loss of loop water flow must deactivate all stages of supplementary heating or heat rejection. When flow resumes, a time delay between stages to limit current in-rush should follow system flow proof. An audible signal should be activated and maintained, even with automatic pump sequencing, to indicate required maintenance.

Chapter 6. Control of heat pump units

A. Control objectives

Space temperature controls for the terminal units will generally include low voltage wall thermostats, although console type (under window) units may accommodate unit mounted controls.

Flexibility of control arrangement to meet any requirement, high reliability, and low cost represent the principal advantages of electric space temperature control systems.

Commercial or office type applications and schools are usually specified with automatic changeover. Apartments, motels and nursing homes usually utilize manual heating or cooling selection. Some models may offer manual or automatic fan speed selection.

Most zones will require one unit with one thermostat, but most buildings contain at least one large zone where two or more units would condition the space more effectively, with one or more thermostat. These zones require a cautious approach, since serious problems will result from improper control system design.

A zone with two or more automatic thermostats must have the thermostats far enough apart to negate the tendency of the units to "fight" each other, wasting energy, and producing poor temperature control. A preferred arrangement involves one thermostat controlling two or more units, either simultaneously or staged.

Thermostat cycling should never reset units after operating fault conditions cause a lockout. Reset at the disconnect switch or circuit breaker provides the only acceptable method of protecting the equipment and the building from possible damage due to resetting equipment not in operating condition. The power disconnect method of reset inherently provides for involvement of maintenance or service personnel. Thermostat reset allows inadvertent reset through changing load requirements in the conditioned space, permits routine reset by unqualified persons, and generally compounds minor service problems into major repairs.

Multiple unit systems should include isolation relays to prevent feedback and loss of control in the event of single unit failure. Avoid control circuit transformer phasing and paralleling. Available isolation relay packages permit one wall thermostat to control an unlimited number of terminal units.

Console type units with integral unit mounted controls are furnished with a constant fan arrangement. Occasionally, this will produce a user complaint about the "cold" air circulated in the heating mode during thermostat satisfaction. Field adaptation of these units to provide cycled fan will result in undesirable and vastly wide differences in space temperature. The thermostat residence within the unit cabinet will isolate it from its controlled space if air circulation ceases.

Public area thermostats should include locking devices to prevent tampering, and some areas may require guards to provide physical protection.

B. Thermostat sensitivity

The design of low voltage wall thermostats, as typically available from Honeywell, Robert-Shaw and White-Rodgers, incorporates the proper switching differentials, heating and cooling anticipators, and response rates for optimum space temperature control within the terminal unit limitations.

C. Night setback

Night setback apparatus, the "best buy" of any system option, reduces energy consumption more than any other feature, costs less to add to the system, and provides a complete return on investment in the first year of system operation.

Individual zones may propel the approach to implementing night setback, typically exemplified by an electronic programmable type controller. An alternative system involves central integration of zone control.

Heat pump manufacturers offer a wide variety of circuits and apparatus whose complexity prevents detailed methodological analysis here. The important thing is what they do, not how they do it. In general, these systems include an overriding control arrangement which, in conjunction with one or more centrally located programming clocks, accomplishes the following:

1. Restart all air conditioners after a general shutdown, from a central point, when so desired.

2. Stop all air conditioners from the same central point when desired.

3. Restart of the air conditions, as in (2) above, should occur in random sequence to limit the instantaneous current demand to a reasonable minimum.

4. Keep all electric circuits to all air conditioners energized, at all times, to maintain a minimum conditioned space temperature (night setting).

5. For systems with inherent or occupant selected continuous fan operation, the control system should switch to cycled fan operation during the night shutdown period.

6. Permit manual, timed override of stopped air conditioners to permit occupants, working at other than normal hours, to restart their respective units for a specified period of normal operation.

7. Switch ventilation systems off, and switch corridor lighting to night requirements.

Night thermostats in each perimeter zone may provide minimum space temperature control, or a thermostat may control a group of units (not as energy efficient). Interior zone units need only be switched off at night.

The terminal unit's minimum entering air operating limit (at the entering water temperature of the loop low limit) demands special consideration. The units will not successfully start at a lower temperature provided by night thermostat control.

Time clock programming also depends upon morning warm-up time at design heating conditioning.

Three different types of night setback control systems exist. Each type serves a particular market application and the choice of which type to use is a judgment decision for the system designer.

In general:

1. NSB affords a solid method of annual power consumption reduction for most systems. Typical reductions of 14% to 16% will readily pay for the extra apparatus in one year or less.

2. NSB may increase the electrical demand, since there is no electrical diversity during morning warm-up (or cooldown).

3. Determination of the optimum night temperature for maximum energy conservation involves compromises based on the consideration of:

   a) Heat pump startup and run temperature limits.

   b) Unoccupied period duration.

   c) Thermal storage factor of the building and contents.

   d) Outdoor temperature swing.

4. Computer simulations, corroborated by field experience and other independent tests[1], indicate that the longer the building is unoccupied, the lower the heating setpoint should be (or the higher the maximum setup temperature should be). However, too low a night temperature setting can actually waste energy saved during the night by inducing conditions where the heat pumps run at low COPs during the warm-up period.

5. Systems provide for night setback 8°F -.0°F, +2.0°F (4.4°C -.0°C, +1.1°C) below the daytime heating setpoint for all units with *low voltage wall thermotstats*. The thermostat scale range is 50°F to 95°F (10°C to 35°C). If both water and ambient air fall below 60°F (15.6°C), it is necessary to elevate the water temperature 1°F (0.6°C) for each degree the air temperature is below 60°F (15.6°C) for a successful restart (as in new building system startup).

   This is not recommended as routine procedure and is limited to 40°F (4.4°C) minimum entering air with 80°F (26.7°C) maximum entering water. Apparatus to automatically accomplish startup with air temperatures below 60°F (15.6°C) would be prohibitively expensive, and the system would waste energy operating under these conditions.

6. Console units, which have unit mounted thermostats, have a night temperature sensor set to control at 60°F (15.6°C). Floor level return air to these units, naturally cooler than at the wall thermostat location, results in similar room temperature maintenance.

7. The amount of time required for morning pull-up varies with outdoor temperature and duration of the night (unoccupied) period.

8. All McQuay International NSB systems provide individual zone control, wasting no energy by operating a unit when its zone does not require heating or cooling, and the after hours override energizes only units actually required. NSB systems are generally not applicable to hospitals, nursing homes, etc.

9. NSB systems for apartments, hotels, motels, and some nursing homes require individual programming of each zone. Central programming may serve certain portions of these buildings.

10. A central system generally controls office buildings, department stores and factories, although different areas of these buildings may be programmed on different schedules.

11. Reverse logic may better serve some zones with unpredictable occupancy, such as student recreation areas. The normal control mode is "unoccupied," and an auto-reset timed override device is used to obtain normal temperatures any time the space is used.

12. During the morning pull-up period, most of the terminal units operate in the same mode. In winter, the perimeter units operate in the heating mode, while interior zone units are off. In summer, all units operate in the cooling mode. The provision of storage tanks and/or special loop water system controls will limit electrical demand and reduce usage rates.

    a) A low temperature tank stores day surplus heat energy for overnight use.

    b) A high temperature storage system permits purchasing supplementary heat requirements at off-peak and/or low rate times.

    c) A chilled water tank, or special evaporative cooler controls, rejects surplus heat at off-peak rates and effects more efficient operation of cooling units the following day.

13. Pneumatic systems should not be used. They offer no advantage, and present several serious disadvantages such as high installed cost, poor control, and poor system reliability.

[1] See ASHRAE Journal 12/75, page 67.

D. Economizer cycle

An economizer cycle could be incorporated into some or all of the individual heat pump units. This economizer cycle would commonly provide for space cooling via outside air during periods when outdoor conditions, temperature and humidity, permit the use of ventilation air to satisfy the room cooling load.

Ostensibly, an economizer cycle conserves energy by using outside air for cooling in lieu of operating the compressor.

In actual practice, however, an economizer cycle will waste energy under certain operational conditions of a closed loop water source heat pump system. For example, during periods when the perimeter zone heat pumps operate in the heating mode, they operate with a coefficient of performance (COP) of approximately 3 as long as their heat of absorption consists of energy stored in the water loop, or rejected into the water loop by interior zone units operating in the cooling mode.

After exhaustion of the stored energy supply, continued perimeter zone heating depends on the transfer of surplus energy from interior zones requiring cooling. If heat pumps operating in the cooling mode accomplish the interior zone cooling, their heat of rejection (to the water loop) becomes a heat source for the perimeter units.

If heat pumps operating in economizer cooling mode accomplish the interior zone cooling, no heat rejects to the water loop, forcing the energization of a central boiler *or* an electric resistance air heater to permit further operation of the perimeter zone heat pump units. This condition allows the realization of a coefficient of performance of only 1.0 during perimeter zone heating.

It is therefore more economical to transfer surplus energy from core areas via the closed water loop than it would be to utilize a boiler as a supplementary heat source whenever the perimeter units operate in the heating mode.

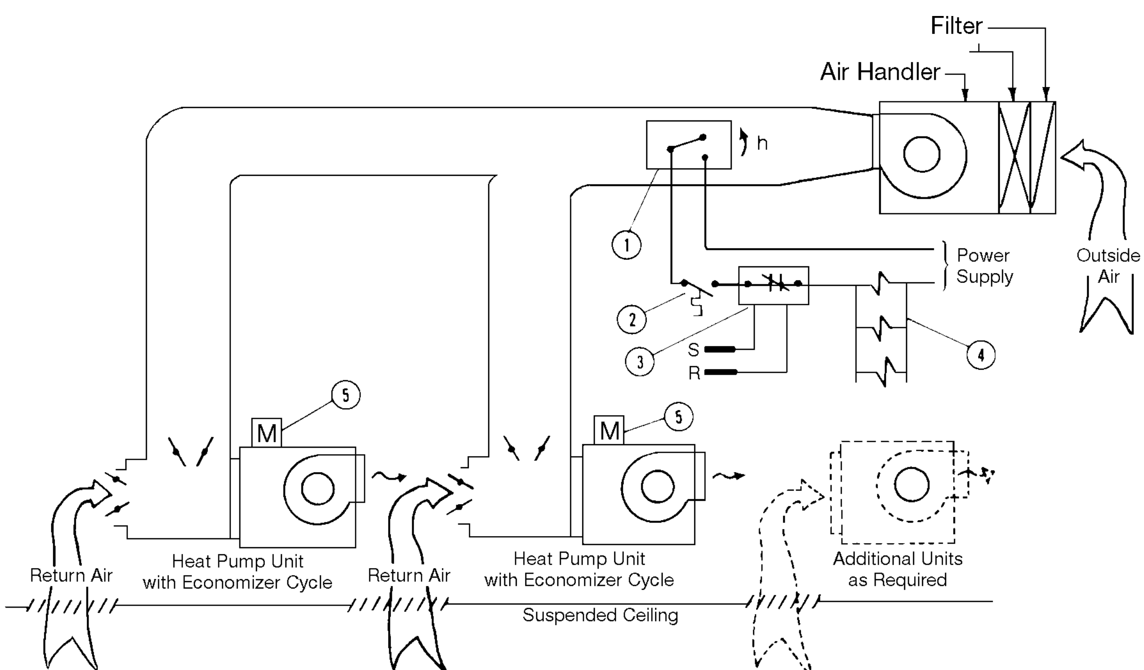
Conversely, the core area economizer cycle should be initiated during perimeter zone unit cooling mode operation. *This will reduce the rate of loop temperature increase, and delay the energization of the heat rejecter device.

***Note:* It is assumed that the cost of economizer cycle apparatus and controls precludes their use in small console or other perimeter zone units. Actually, all units could have economizer cycles, but the principle of operation remains the same: the economizer cycles must be locked out under certain circumstances to transfer energy within the building in preference to purchasing new energy.**

McQuay International can provide a control logic scheme which eliminates the discarding of interior zone energy when perimeter units could make use of that energy. This logic scheme can also provide a low loop water temperature limit below which economizer operation is prohibited.

Figure 1 illustrates a typical approach for providing a central control system for a number of heat pump units, allowing the use of inexpensive spring return economizer damper motors.

Figure 1. Low cost economizer system



1. Enthalpy Control.
2. Loop Water Low Limit T'Stat 70°F (21°C).
3. Loop Water Supply/Return ΔT Control.
4. Economizer Relays.
5. Economizer Damper Motor. Two Position Spring Return.

A series circuit consisting of ① an enthalpy controller (or thermostat) to qualify outside ventilation air as suitable for cooling, ② a loop return water temperature sensor, and ③ a differential thermostat with loop water temperature sensors forms the essence of the arrangement. One sensor is installed in the return loop water line as shown in Figure 2 ⑤, and the other sensor is installed in the supply loop water line, Figure 2 ≈ . The differential thermostat monitors the temperature difference between supply and return water, and permits economizer operation only when the return water temperature exceeds the supply temperature. Under circumstances where the heat absorbed by units heating nearly balances the heat rejected by units cooling, making the temperature difference between supply and return loop water less than the differential thermostat sensitivity, the return water temperature sensor ② prohibits economizer operation.

This process only occurs if a gradual drop in loop temperature indicates the building heating requirement exceeds the building cooling requirement.

The economizer relay √ in Figure 1 can be a multiple pole type, or multiple relays with the coils paralleled, for control of any number of units.

The thermostat ∆ in Figure 3 with two-stage cooling provides for compressor operation and supplemental refrigeration cycle cooling when the economizer cycle is incapable of satisfying the cooling load.

The night setback relay ⑧, Figure 3, is energized at night to (a) lock out cooling, (b) switch heating control point from the desired "room occupied" value to a lower "room unoccupied" value, and (c) switch the fan from constant to cycled operation.

Figure 2.

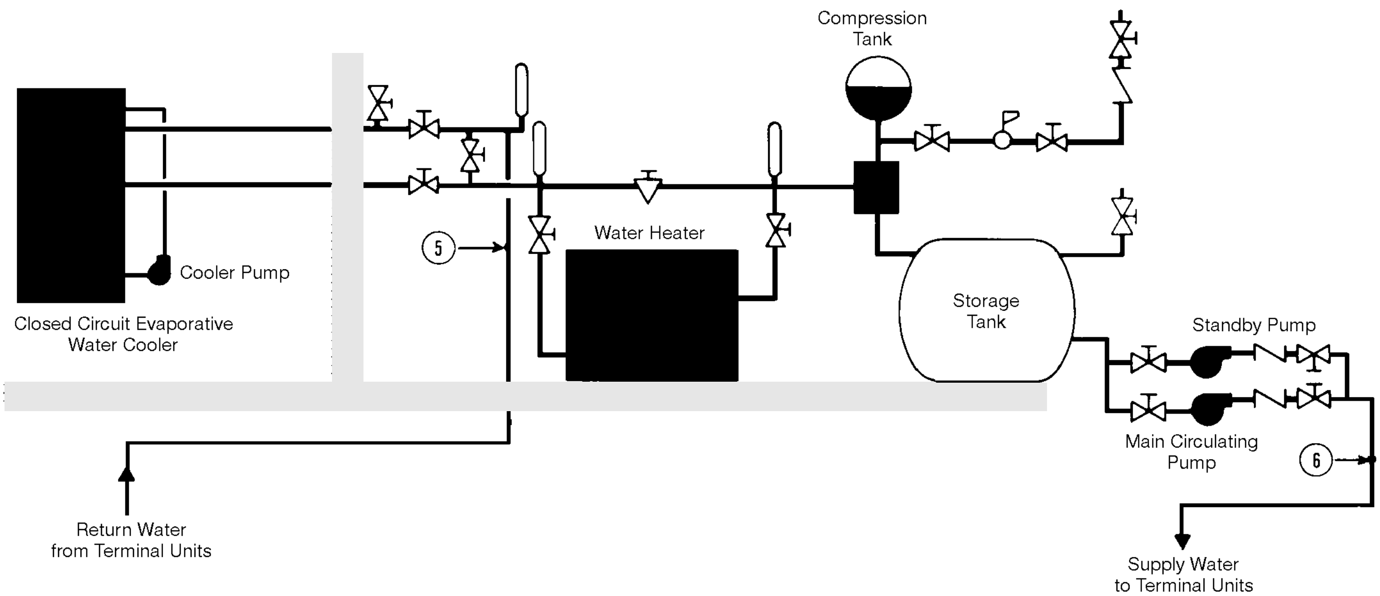
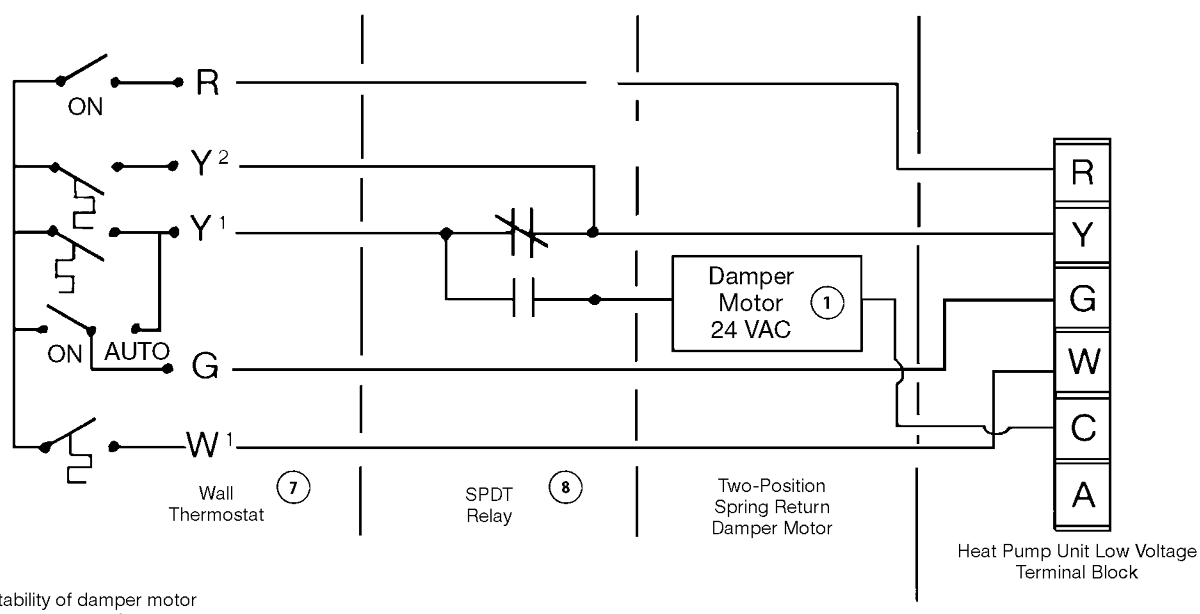


Figure 3.

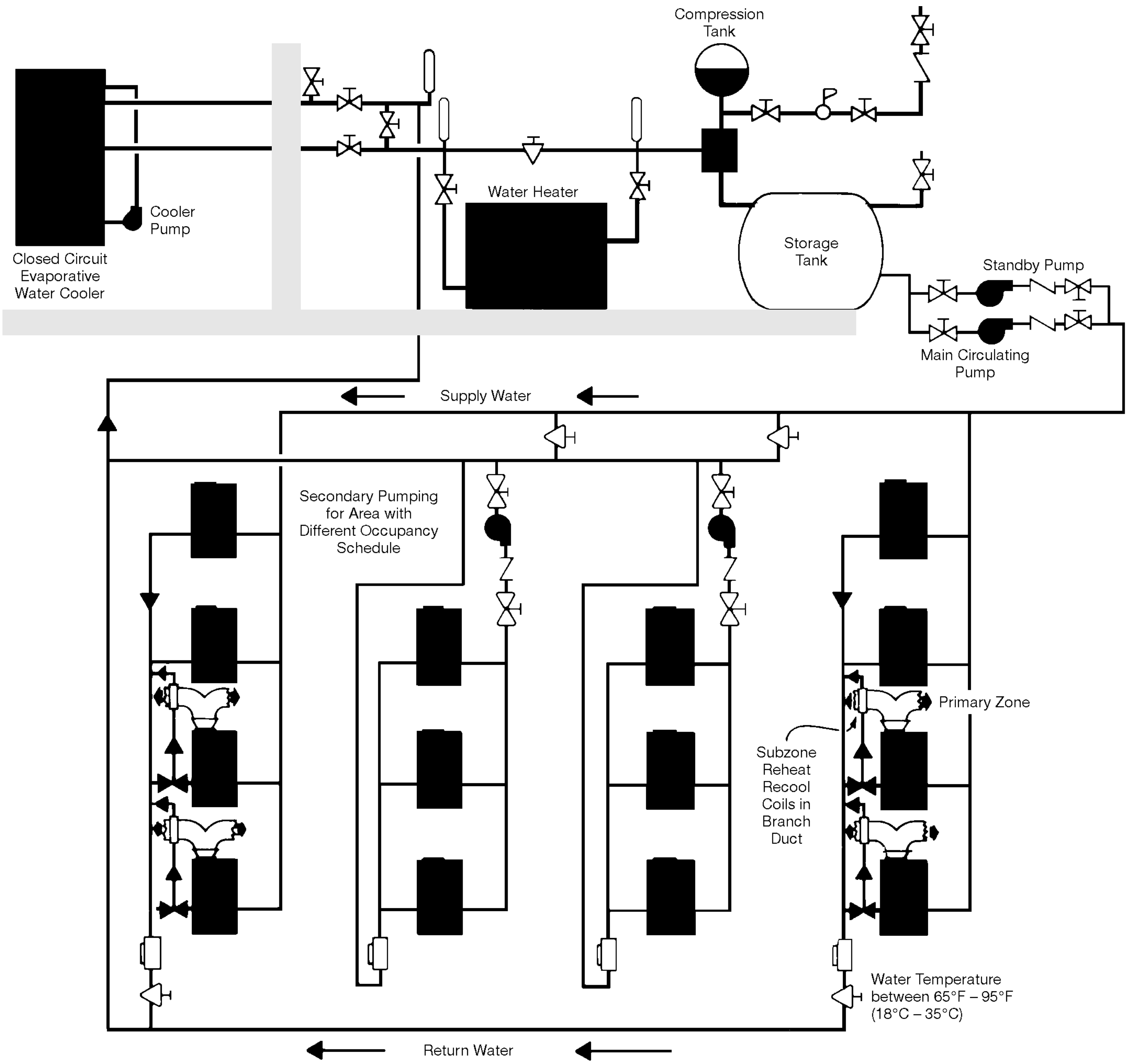


① Verify acceptability of damper motor load with heat pump manufacturer.

Chapter 7. Miscellaneous design considerations

A. Primary/secondary pumping. For buildings with areas subject to different time cycle occupancies — perimeter offices, interior zone areas, or restaurants, etc. — a primary/secondary pumping system can achieve hydraulic isolation of each secondary circuit. This system may cut off flow in a secondary circuit (e.g. the core) along with the associated heat pumps during unoccupied periods to maximize operating cost savings. Shutting off such a secondary pumping loop would not disturb the system hydraulics.

Water source heat pump system with primary/secondary piping arrangement



B. Individual units do not require **air vents**, because any air will be entrained except when encountering water velocities low enough to preclude heat pump operation. Vent only the system high points.

C. A 1750 rpm **direct drive centrifugal pump** is generally preferred over a 3450 rpm pump for quiet operation. This pump should be selected for a mid-curve operating condition.

D. Size the **expansion tank** for a 50°F to 110°F (10°C to 43.3°C) temperature range, approximately 2% of total volume of water in the system.

E. **Pressure gauges** across strainers are usually worthwhile.

F. Copper and iron pipe are compatible on closed system, no air.

G. The minimal scaling of water on the coil of the evaporative water cooler may be ignored. Tube surface temperature only reaches 105°F (40.6°C) maximum versus the 140°F (60°C) customary with evaporative condensers. Total annual operating time also decreases considerably due to energy conservation.

H. Consider swimming pool heaters for smaller systems in lieu of regular boilers.

I. A direct return system with flow control devices can function well, but these devices require careful selection and thorough system flushing prior to installation.

J. Wherever employing a water-to-water heat exchanger, ensure that loop water connects to the shell side, thus allowing cleaning access to the tube side, which tends to foul.

K. Consider placing the domestic hot water in heat exchange with loop water to achieve a 10°F to 20°F (5.6°C to 11.1°C) rise before entering the domestic water heater. In most applications, this represents a "free" heat source.

L. Observe manufacturer's product application limitations and solicit the manufacturer's recommendations for peculiar situations. For example, do not use equipment designed for indoor use in an uninsulated attic.

M. A water source heat pump system will likely incorporate exposed outdoor closed water loop components. Examples might include the coil of a closed circuit evaporative water cooler, or the water-refrigerant heat exchanger of a roof mounted heat pump. In climates where freezing temperatures can be encountered, a prolonged winter power failure could cause damage if the water freezes within one of these components.

In systems invoking designer concern about the possibility of freezing any part of the water loop, a 10% by volume inhibited anti freeze solution is recommended. Avoid higher concentrations than this. A 10% or smaller concentration has a negligible effect on system performance, but performance deteriorates with greater concentrations. For concentrations between 35% and 50%, the lower solution specific heat requires an increased flow rate. The resultant increased friction caused by *both* the greater flow rate and higher fluid viscosity drastically increases pumping power. The reduced thermal conductivity of glycol solutions also induces a significant penalty in heat pump unit performance due to their lower heat transfer coefficient.

Although a 10% solution of anti freeze will not totally prevent freezing at temperatures below 24°F (-4°C), it has demonstrated effectiveness in preventing physical damage to metal piping and heat exchangers of the closed water loop at any temperature. For conditions conducive to freezing, a slush forms with the development of ice crystals. The remaining liquid has a higher glycol concentration. As the icy slush forms, expansion occurs, but the pipe loop's compressor tank within the building absorbs this expansion. Given sufficient time and cold temperature, the solution can eventually solidify, but with a consistency similar to crystalline ice cream, and only after the completion of the expansion which occurs with freezing during the slush stage. Serpentine coils and piping constructed of materials such as copper or iron have not broken or distorted due to this type of freezing during laboratory tests, nor has such damage been reported in field installations.

A 10% solution adequately protects the water source heat pump system because of the design of this system and its component products. Coils through which the system water circulates are serpentine to avoid expansion restriction within the coil. No automatic valves which might close to totally segregate or isolate any component that might suffer exposure to freezing conditions exist. Other systems that do have automatic valve control could restrict the expansion occurring during slush formation, with the possibility of rupturing some portion of the isolated circuit. Similarly, non-serpentine coils might prompt damage to headers if expansion was restricted by slush forming a "dam" at the entry to smaller tubes leaving the header.

Because of the importance of inhibitors, avoid automotive anti freeze formulations. The inhibitors they contain may react adversely with other materials present, or with other inhibitors subsequently added to maintain the necessary inhibitor concentration. Instead, use an industrial formulation such as DOWTHERM SR-1 by Dow Chemical Company, or UCAR Thermofluid 17 by Union Carbide Corporation. Both products utilize dipotassium hydrogen phosphate (K_2HPO_4) and "NaCap" copper deactivator as inhibitors. Because of the low ethylene glycol concentration recommended for this application, the amount of inhibitor included with the glycol will not be adequate. It is necessary, therefore, to add additional inhibitors simultaneous with ethylene glycol introduction into the pipe loop. Refer to Chapter 4 covering water treatment of the water source heat pump system.

N. **An alternate approach to evaporative cooler selection,** (where the cooler manufacturer does not base data on total connected horsepower versus outside design wet-bulb temperature) involves determining:

1. The amount of heat to be rejected
2. The outside design wet-bulb temperature
3. The range (temperature difference between water entering and leaving the cooler)
4. The approach (temperature difference between water leaving the cooler and the outside design wet-bulb)
5. Optimum system flow rate

Note: In an *evaporative cooler*, the outside dry-bulb temperature does not have relevance. The most difficult concepts to grasp involve the effect of flow rate on annual system operating cost, the temperature range through which the water may fluctuate, and their interrelationship.

The terminal heat pumps, operated in the cooling mode, have an optimum condensing temperature at which maximum cooling capacity production transpires. Higher condensing temperatures result in reduced cooling capacity and increased power consumption. However, good design practice does not encompass an attempt to provide a flow rate or cooler selection which will maintain terminal unit operation at maximum efficiency. Instead, terminal unit condensing temperature is permitted to rise until the amount of energy required to reject surplus heat equals terminal unit losses which would otherwise result upon the allowance of further loop temperature rise.

Use of the same logic, in reverse sequence, can determine the optimum energization point for the loop supplementary water, except that outdoor reset controls may be applied to vary the minimum loop temperature in reverse proportion to the outdoor ambient.

The system flow rate also affects the loop temperature control points. A high flow rate of warm water may provide an acceptable heat sink for terminal units operating in the cooling mode, but the same loop water must act as a heat source if certain zones require heating. For terminal units operating in heating, a high flow rate of warm water could cause unit lockout on compressor thermal protection or high pressure. The circulation of loop water at the high flow rate also requires more pumping power, which on an annualized basis, wastes energy.

A low flow rate saves pumping power but also narrows the temperature range through which the loop water can fluctuate without energizing either the heat rejecter or the heat adder (boiler), squandering system energy conservation potential.

To avoid safety lockouts and energy misuse, operate units only within manufacturer-recommended flow rates. McQuay International has developed optimum system design criteria considering:

a) System annual operating hours at design conditions 5% of the time, and at less than half load 75% of the time.

b) Terminal unit maximum condensing temperature (cool)

c) Terminal unit minimum condensing temperature (cool)

d) Terminal unit maximum evaporator load (heat)

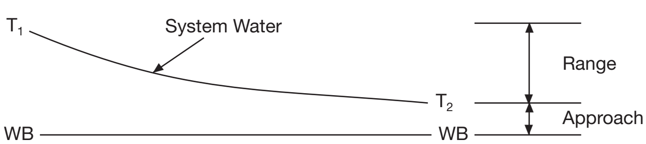
e) Terminal unit minimum evaporator lead (heat)

f) Most systems require any heat pump terminal to heat or cool at any time.

g) Relative power to operate terminal units, circulating pump and heat rejecter.

h) The outside design wet bulb temperature

Knowing the design wet bulb temperature for your area, simply enter the table and read cooling range, approach and flow rate per total connected load.



Range, T_1-T_2 = 9.5°F (5.3°C)
Approach, T_2 - WB = 14.5°F (8.1°C)
T_2 = WB + approach = 78 + 14.5 = 92.5°F
= 25.6 + 8.1 = 33.6°C
T_1 = T_2 + range = 92.5 + 9.5 = 102°F
= 33.6 + 5.3 = 38.9°C
gpm/hp = 2.45
L/s/kW = 0.044

Outside Design W.B. °F (°C)	Temperature Leaving The Cooler °F (°C)	Flow Rate GPM/Ton (L/s/kW)	Cooler Range @ 75% Diversity °F (°C)	Approach °F (°C)
65 (18.3)	90.0 (32.2)	2.00 (0.036)	11.3 (6.3)	25.0 (13.9)
66 (18.9)	90.0 (32.2)	2.00 (0.036)	11.3 (6.3)	24.0 (13.3)
67 (19.4)	90.0 (32.2)	2.00 (0.036)	11.3 (6.3)	23.0 (12.8)
68 (20.0)	90.0 (32.2)	2.00 (0.036)	11.3 (6.3)	22.0 (12.2)
69 (20.6)	90.0 (32.2)	2.00 (0.036)	11.3 (6.3)	21.0 (11.7)
70 (21.1)	90.0 (32.2)	2.00 (0.036)	11.3 (6.3)	20.0 (11.1)
71 (21.7)	90.0 (32.2)	2.00 (0.036)	11.3 (6.3)	19.0 (10.6)
72 (22.2)	90.0 (32.2)	2.00 (0.036)	11.3 (6.3)	18.0 (10.0)
73 (22.8)	90.0 (32.2)	2.00 (0.036)	11.3 (6.3)	17.0 (9.4)
74 (23.3)	90.0 (32.2)	2.04 (0.037)	11.3 (6.3)	17.0 (9.4)
75 (23.9)	91.0 (32.8)	2.19 (0.039)	10.6 (5.9)	16.0 (8.9)
76 (24.4)	91.5 (33.1)	2.27 (0.041)	10.2 (5.7)	15.5 (8.6)
77 (25.0)	92.0 (33.3)	2.36 (0.043)	9.8 (5.4)	15.0 (8.3)
78 (25.6)	92.5 (33.6)	2.45 (0.044)	9.5 (5.3)	14.5 (8.1)
79 (26.1)	93.0 (33.9)	2.55 (0.046)	9.1 (5.1)	14.0 (7.8)
80 (26.7)	93.5 (34.2)	2.66 (0.048)	8.7 (4.8)	13.5 (7.5)
81 (27.2)	94.0 (34.4)	2.78 (0.050)	8.3 (4.6)	13.0 (7.2)
82 (27.8)	94.5 (34.7)	2.91 (0.052)	8.0 (4.4)	12.5 (6.9)

Thus a system with 33 heat pumps, whose ARI total cooling capacity rating is 52,000 Btuh (15.2 kW) each, would circulate 350 gpm (22.1 L/s), entering the cooler at 102°F (38.9°C), and leaving the cooler at 92.5°F (33.6°C), at 78°F (25.6°C) WB.

Note the consideration of diversity in the range. The example demonstrates a 75% diversity factor, which represents 86.6% of the terminal units running 86.6% of the time, at design conditions. System diversity is *never* 100%. The range through any operating individual terminal unit is 9.5°F (5.3°C) ÷ 0.75 or 12.7°F (7.1°C). If the cooler were selected for 70% diversity (equal to 84% of the units operating 84% of the time), the range through the terminal units would remain 12.7°F (7.1°C). The flow rate would not change, but the range value used for cooler selection would be 8.9°F (4.9°C) which would be a smaller cooler.

Cooler selection can involve the supposition of significant diversity factors. Too many systems have oversized coolers which are not cost beneficial. High first costs, significant stress to building structure due to extra weight, and sacrificed operational economy all result from the selection of an overly large cooler. A prudent designer will devote thorough analysis to this subject, considering average summer temperatures, the frequency and duration of design conditions, the proportion of the load not subject to diversity (computer cooling, etc.), type of occupancy and time of day during which the load occurs, etc.

O. High altitude applications — Fan tables and curves are based on air at standard atmospheric conditions of 70°F (21.1°C) and 29.92 in. Hg (760mm Hg) barometric pressure. If a fan must operate at nonstandard conditions, the selection procedure must include a correction. With a given capacity and static pressure at operating conditions, make adjustments as follows:

1. For packaged units with adjustable belt drive fans:

a) Obtain the air density ratio from Table 1.

b) Calculate the equivalent static pressure by dividing the given static pressure by the air density ratio.

c) Enter the fan tables for the unit at the given capacity and the equivalent static pressure to obtain speed and brake horsepower. This speed is correct as determined.

d) Multiply the tabular brake horsepower by the air density ratio to find the brake horsepower at the operating conditions.

2. The air delivery of units with direct drive fans will decrease with increases in altitude and/or increased external static pressure.

   To determine the air delivered by a direct drive fan:

   a) Obtain the air density ratio from Table 1.

   b) Calculate the equivalent static pressure by dividing the given static pressure by the air density ratio.

   c) Enter the fan tables for the unit at the equivalent static pressure to obtain the actual air delivered at the operating conditions.

   d) Ensure that the actual air delivered is not less than the minimum airflow specified by the heat pump manufacturer.

   e) Calculate the effect of actual air delivery on unit performance by applying the proper multiplier to catalog rating values. Percent of rated flow rate is obtained by dividing the actual flow rate at operating conditions by the catalog rating flow rate.

P. Winter cooler bypass — Bypassing the outdoor evaporative water cooler to prevent winter heat loss from the closed water loop may prove worthwhile in certain system designs. **Do not incorporate** cooler bypass in installations which reject heat during occupied hours on an "average" winter day. The employment of bypass valving and controls on such installations reduces system reliability and seldom provides cost benefits.

If the system interior zone cooling load is small in relationship to the perimeter zone heating load, bypassing the evaporative water cooler is prudent. However:

1. Manufacturers discourage the use of manual valving. Experience has shown that operating personnel frequently fail to re-establish the cooler in the circuit when the inevitable, odd warm winter day occurs.

2. Freezing presents a threat to the isolated fluid in the cooler. Draining permits coil corrosion to occur and dilutes system water treatment when refilled.

   An anti-freeze solution protects the isolated cooler coil from freeze-up, but imposes an annual operating penalty on the entire system due to the higher viscosity of the solution and poorer heat transfer characteristics.

3. A two-position, three-way automatic bypass valve is not recommended unless using a "slow-acting" operator. The use of such a valve results in a "slug" of cold water injected into the system upon re-establishment of flow through the cooler. The thermal response rate of system controls may induce supplementary water heater energization, and as the "slug" of cold water circulates to the terminal air conditioning machines, they may lock out on operating safety switches.

   Where employing cooler bypass and providing proper freeze-up protection (anti-freeze or cooler coil draining) one or more automatic, modulating three-way valves are preferred. Slow acting two-position valves probably provide an acceptable alternative where employing multiple valves.

Important: All temperature sensor system controls and the system flow switch must not be installed in the portion of the loop associated with the bypassed cooler.

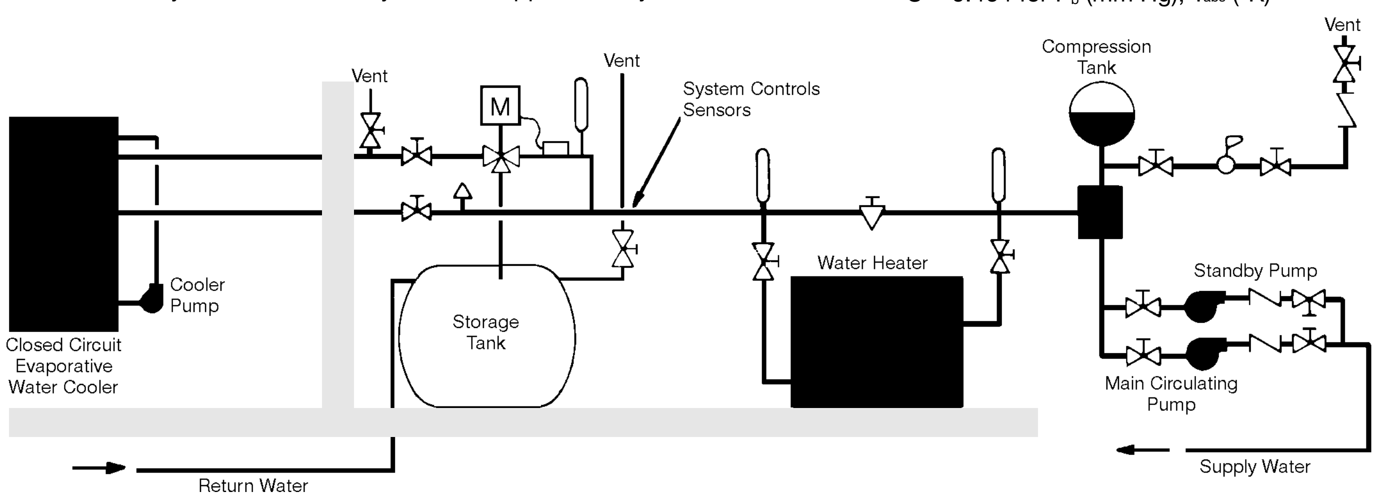
Table 1.

Altitude Ft. (m)	Pressure (P_b) In. Hg (mm Hg)	Density (d)* Lb/Ft^3 (kg/m^3)	Density Ratio
0 (0)	29.92 (760)	0.0748 (1.198)	1.000
500 (152.4)	29.38 (746)	0.0735 (1.177)	.982
1000 (304.8)	28.86 (733)	0.0722 (1.157)	.965
1500 (457.2)	28.34 (720)	0.0709 (1.136)	.948
2000 (609.6)	27.82 (707)	0.0696 (1.115)	.930
2500 (762)	27.32 (694)	0.0683 (1.094)	.914
3000 (914.4)	26.62 (681)	0.0671 (1.075)	.896
3500 (1066.8)	26.33 (669)	0.0659 (1.056)	.881
4000 (1219.2)	25.84 (656)	0.0646 (1.035)	.864
4500 (1371.6)	25.37 (644)	0.0635 (1.017)	.848
5000 (1524)	24.90 (632)	0.0623 (0.998)	.832
5500 (1676.4)	24.43 (621)	0.0611 (0.979)	.817
6000 (1828.8)	23.98 (609)	0.0600 (0.961)	.802
6500 (1981.2)	23.53 (598)	0.0589 (0.943)	.787
7000 (2133.6)	23.09 (586)	0.0578 (0.926)	.772
7500 (2286)	22.60 (574)	0.0565 (0.905)	.756
8000 (2438.4)	22.22 (564)	0.0556 (0.891)	.743

*Density @ 70°F (21.1°C), $d = \frac{C\,P_b}{T_{abs}}$

$C = 1.325$ for P_b (in. Hg), T_{abs} (°R)

$C = 0.464$ for P_b (mm Hg), T_{abs} (°K)



Q. Circulating pump control — Turning off the system pumps when the conditioned space does not require heating or cooling allows the realization of significant energy savings. The loop water pump power generally comprises approximately 6.5% of total system power. Turning off the pumps during quiescent system periods effects pump power savings of approximately 30%.

Before committing the fundamental error of simply discontinuing power to the pumps, designers must provide a control scheme to:

1. Interlock heat pump operation to prohibit operation during lack of pump performance (or else the units will routinely cycle on their safety switches, causing premature failures and voiding most manufacturers' warranties).

2. Automatically restart the pumps (override the pump off schedule) whenever:

   a) An interior override thermostat, placed in the least favored location, indicates that the maintenance of minimum space temperatures will require night setback; or maintenance of maximum temperature through cooling operation (night setup).

   b) An outside override thermostat indicates that a period of low outside ambient requires pump operation to prevent freeze-up in outdoor portions of the loop, particularly the cooler.

3. Provide a time delay of several minutes between pump startup and heat pump unit startup. Portions of the loop may lose heat or gain excessive heat while the pumps do not operate. On startup, these "slugs" of cold or warm water would enter the heat pumps and generate nuisance trip outs. The time delay "conditions" the water to defeat this probability.

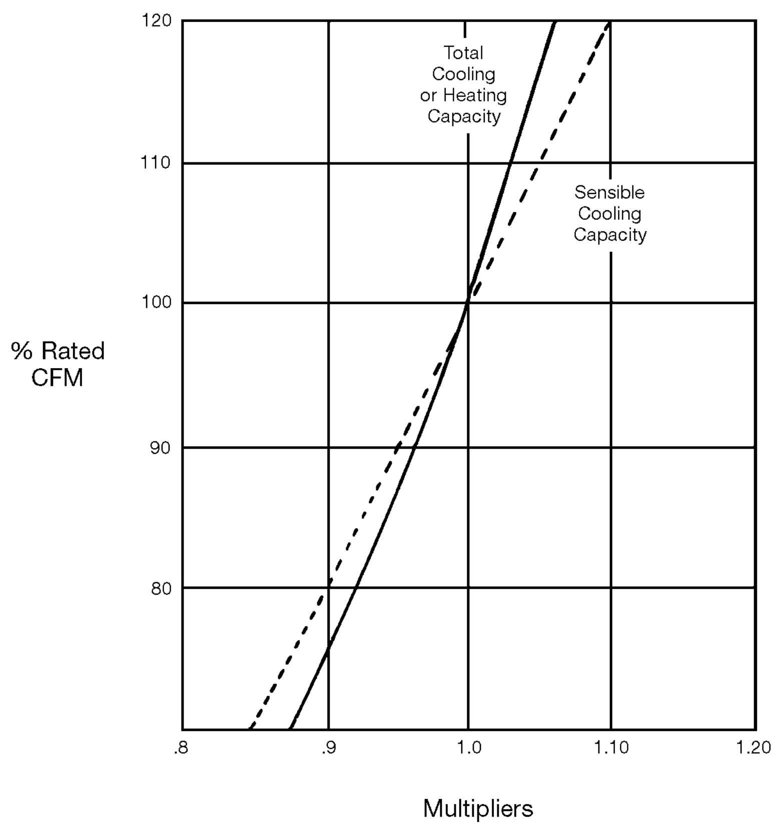
R. Variable volume pumping — McQuay International has field proven a method of modulating the pumps to provide only the flow required for operating heat pump units. This is a major advancement in the state of the art.

Pump modulation precludes any further concept or competitive preoccupation with "the big circulating pumps run all the time." Variable volume pumping takes the next major step: system flow actually matches system requirements.

Just as simply as the main blower of a VAV system modulates in response to a signal from the pressure sensor in the discharge duct when the terminals throttle open or closed, the water source heat pumps vary their speed to only circulate the quantity of water actually required.

McQuay International has developed a dual-acting water regulating valve which modulates water flow to the optimum rate when the compressor runs, and discontinues flow when the compressor idles. Thus, variable volume pumping optimizes heat pump performance throughout the loop water temperature range in addition to invoking cost savings. At this time, dual-acting valves only come in ½" and ¾" sizes for the smaller units. Larger units are fitted with a motorized valve.

Very large tonnage units and core cooling only units generally have no valves, allowing the achievement of the objective of 60% to 80% total water flow modulation.



Chapter 8. Building system design worksheet

Load Calculations	Loop Water Flow
1. Building block cooling load has been computed to be: ________________ btu/hr	1. ___________ type ___________units hp (kW) ________ gpm (L/s) ea = _________
2. Building block heating load has been computed to be: ________________ btu/hr	2. ___________ type ___________units hp (kW) ________ gpm (L/s) ea = _________
3. Electric resistance heat has been used to preheat air (not a load on the loop water) in the amount: ________________ btu/hr	3. ___________ type ___________units hp (kW) ________ gpm (L/s) ea = _________
4. Supplemental resistance electric heat has been used to offset glass radiation (not a load on the loop water) in the amount of: ________________ btu/hr	4. ___________ type ___________units hp (kW) ________ gpm (L/s) ea = _________
5. Maximum net required amount of heat that must be supplied to building at worst condition by heat pumps and loop water items [2 – (3 + 4)]: ________________ btu/hr	5. ___________ type ___________units hp (kW) ________ gpm (L/s) ea = _________
6. Amount of heat supplied to loop: **Note:** If there is night set back supply total heat of absorption otherwise 70% of item 5. ________________ btu/hr	6. ___________ type ___________units hp (kW) ________ gpm (L/s) ea = _________
7. kW of heat supplied to loop: Item 6 ÷ 3412 = ________________ kW	7. ___________ type ___________units hp (kW) ________ gpm (L/s) ea = _________
8. Capacity of heat rejector: Enter size of evaporative water cooler, as determined from manufacturer's data. Total connected: hp (kW) __________ unit size ___________ gpm (L/s) _________ gpm/hp (L/s/kW) __________ pressure drop, psi (Kpa) ________	8. ___________ type ___________units hp (kW) ________ gpm (L/s) ea = _________
9. Outdoor summer design wet and dry bulb temperature: _________°F (°C) db ___________°F (°C) wb	9. ___________ type ___________units hp (kW) ________ gpm (L/s) ea = _________
10. Design loop water temperature to system: Maximum = ________________________ Maximum = ________________________	The total gpm (L/s) in the loop water system is the total of the items above, or _________ gpm (L/s). It is advisable, usually, to make all self contained units water cooled and place on loop.

Heat of Rejection	Heat of Absorption
1. ___________ type ___________ units at: ___________ btu/hr (kW) ea = ___________ btu/hr (kW)	1. ___________ type ___________ units at: ___________ btu/hr (kW) ea = ___________ btu/hr (kW)
2. ___________ type ___________ units at: ___________ btu/hr (kW) ea = ___________ btu/hr (kW)	2. ___________ type ___________ units at: ___________ btu/hr (kW) ea = ___________ btu/hr (kW)
3. ___________ type ___________ units at: ___________ btu/hr (kW) ea = ___________ btu/hr (kW)	3. ___________ type ___________ units at: ___________ btu/hr (kW) ea = ___________ btu/hr (kW)
4. ___________ type ___________ units at: ___________ btu/hr (kW) ea = ___________ btu/hr (kW)	4. ___________ type ___________ units at: ___________ btu/hr (kW) ea = ___________ btu/hr (kW)
5. ___________ type ___________ units at: ___________ btu/hr (kW) ea = ___________ btu/hr (kW)	5. ___________ type ___________ units at: ___________ btu/hr (kW) ea = ___________ btu/hr (kW)
6. ___________ type ___________ units at: ___________ btu/hr (kW) ea = ___________ btu/hr (kW)	6. ___________ type ___________ units at: ___________ btu/hr (kW) ea = ___________ btu/hr (kW)
7. ___________ type ___________ units at: ___________ btu/hr (kW) ea = ___________ btu/hr (kW)	7. ___________ type ___________ units at: ___________ btu/hr (kW) ea = ___________ btu/hr (kW)
8. ___________ type ___________ units at: ___________ btu/hr (kW) ea = ___________ btu/hr (kW)	8. ___________ type ___________ units at: ___________ btu/hr (kW) ea = ___________ btu/hr (kW)
9. ___________ type ___________ units at: ___________ btu/hr (kW) ea = ___________ btu/hr (kW)	9. ___________ type ___________ units at: ___________ btu/hr (kW) ea = ___________ btu/hr (kW)
10. ___________ type ___________ units at: ___________ btu/hr (kW) ea = ___________ btu/hr (kW)	10. ___________ type ___________ units at: ___________ btu/hr (kW) ea = ___________ btu/hr (kW)
11. ___________ type ___________ units at: ___________ btu/hr (kW) ea = ___________ btu/hr (kW)	11. ___________ type ___________ units at: ___________ btu/hr (kW) ea = ___________ btu/hr (kW)
The total heat rejection to the loop water with all units on full load (this could occur in some buildings) is _______________ btu/hr (kW).	The total heat rejection to the loop water with all units on full load (this could occur in some buildings) is _______________ btu/hr (kW). **Please Note:** Cooling only units cannot add heat of absorption to the loop.

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